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Hot-Spin Tests of Bladed Jet-Engine Rotors

By H. B. SALDIN[1] AND P. G. DeHUFF, JR.,[2] SOUTH PHILADELPHIA, PA.

The creep-rupture and ductility characteristics of materials are of utmost importance to the designer of high-temperature rotating parts. The authors have pointed out the significance of ductility of materials used for gas-turbine disks. Four-bladed disks were tested in a facility that was designed to spin the rotors in as near engine operating conditions as possible. These parts were tested in accordance with a predetermined schedule of temperature gradient, temperature, and speed. Test results are analyzed, and a discussion of the advantages and the design of such a facility is also included.

Introduction

Creep-rupture tests are of value in determining the life of a given material under variable loads and temperatures. When it comes to evaluating the relative merits of materials suitable for use as jet-engine turbine disks, more complete testing is necessary. To be considered are the stresses due to thermal gradients and rotation; the effect of bolting restrictions and notches; and, related to all of these, the effect of ductility. Of particular significance is this intangible ductility. Most disk materials are very ductile at room temperature, but some are much less ductile at elevated temperature, and some materials have little or no hot ductility after processing. Designers know that many of the currently used materials can be processed to have high strength with low ductility or low strength with high ductility. There has always been uncertainty of the minimum ductility for safety in a given application. One of the main functions of disk testing is to help evaluate this "strength versus ductility" situation.

Careful examination of the possible methods for selecting and testing disk materials revealed that the operation of bladed disks in an engine or under similar conditions would be the best way of combining the many variables into one test.

An analysis was made of the advantages and disadvantages of running bladed disks in an actual engine. The advantages are as follows:

1 Represents identical conditions to normal engine running.
2 No additional equipment needed.

The disadvantages included the following:

1 Optimum personnel safety is difficult and costly.
2 Costly engine damage in case of a wreck.
3 Rotational-speed limitation by other engine parts.
4 High operating cost.
5 Temperature measurements difficult.
6 Design and configuration inflexible.
7 Little independent control of variables.
8 Excessive change-over time.
9 Difficulty in repeating test conditions and results.
10 Measurements of creep are difficult without extensive engine teardown.

A quick glance at the foregoing analysis shows that although a bladed disk could be run in an engine for varying periods of time, there are not many conditions which can be varied during engine operation. Speed cannot be increased without seriously overloading other parts of the engine, particularly the turbine blades. The other items such as high cost and poor control of test conditions are additional cause for looking at a separate hot-spin test pit as the best method of performing the tests.

In 1946 N. L. Mochel (1)[3] discussed the problem of hot ductility in gas-turbine rotors and pointed out the need of evaluating this material property in a hot-spin test pit.

Mr. Fonda (2) has outlined a series of tests in which unbladed disks have been spun cold to destruction. It is the purpose of our hot-spin test pit to spin disks fully bladed and at the temperature of normal engine operation under conditions where disk growth and behavior can be observed accurately up to and including disk failure.

There is another hot-spin disk test pit in operation at the author's company laboratories in East Pittsburgh, which is engaged in the exploration of fundamental disk design including the effect of high temperatures. This series of tests is aimed at correlating theoretical disk calculations with actual tests of unbladed disks. The Aviation Gas Turbine Division tests are primarily for evaluating materials and design for engine application.

Test-Pit Design

Late in 1945, after the Aviation Gas Turbine Laboratory (3) was in existence and an operating branch of the Aviation Gas Turbine Division, it was decided to build a disk test pit for developmental testing.

Among the primary requisites of a test pit are safety, ease of assembly and disassembly, a minimum of parts that need repair or replacing after disk failures, and low power requirements.

During our investigations prior to the design of the pit, we found there was quite a divergence of opinion as to what was adequate safety protection. Our test pit was located on the ground floor of the building where the rotating test disk could be operated entirely below ground level. The hole was lined with 15 in. of reinforced concrete, this being a very easy way of obtaining a large amount of protection from disk explosions. In order that the disk fragments might be stopped in a wall somewhat easier to repair, two walls of $2^1/_4$ in. of laminated steel plate were placed inside the concrete, each wall consisting of 9 layers of $^1/_4$-in. plate. Then in the hope that the disk fragments might be stopped by some material less damaging to the disk fractures, a $4^1/_2$-in. wall of radiused lead brick was laid inside the laminated steel. This was viewed as ample protection with a very large factor of safety to take care of all the contingencies, even including those indicated by reports that would make it seem almost a hopeless task to stop the disk fragments. As clearly indicated in the sectional view of the test pit, Fig. 1, considerably less protection was provided from ricocheted fragments which

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[3] Numbers in parentheses refer to the Bibliography at the end of the paper.

might go up, than in any other direction. This was augmented by an armor-plate cover which was used until experience indicated it was not necessary. The disk failures have proved that our safety precautions were adequate, and that the additional armor over the top was unnecessary. Experience indicates that fragments of the disk penetrate only about $^1/_3$ to $^1/_2$ the thickness of the lead brick, Fig. 2.

The lowest practical absolute pressure is maintained in the test pit to reduce the power to a minimum required to rotate the test disk at high speed. Referring to Fig. 1, there is indica the large steel shell just inside the lead-brick wall which forms vacuum wall for the test pit. This shell is considered semi pendable and holes made from bursting disks are repaired, a the shell used again.

The assembly thus far described constitutes the station elements of the test pit. The complete test assembly is moun on the cover for the vacuum tank, which is easily removed fr the pit by a motor-operated monorail crane. The edge of t

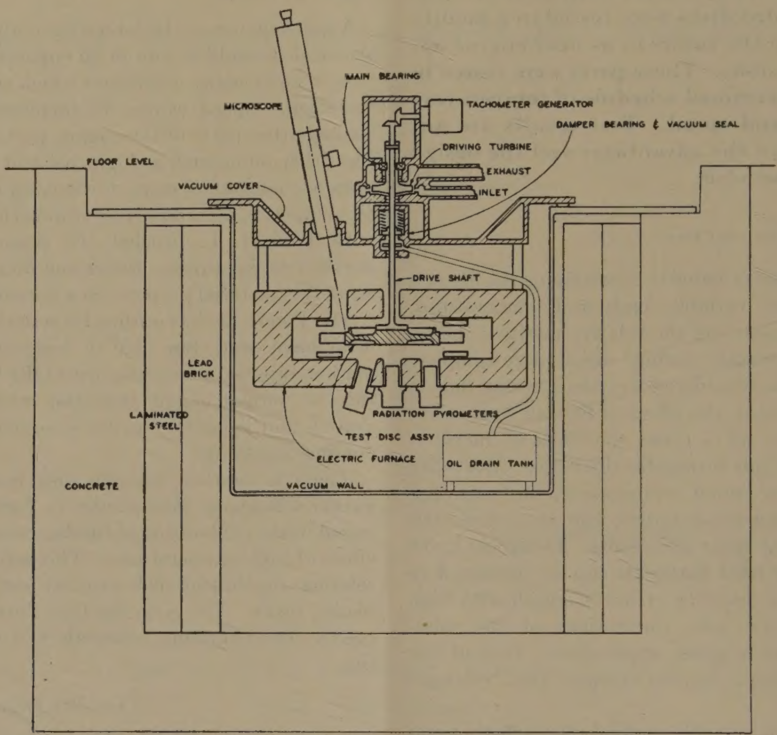

Fig. 1 Longitudinal Section of Hot-Spin Test Pit

Fig. 2 Typical Test Pit Wall Damage

dished cover has a lip which rests on a heavy rubber gasket to make the vacuum seal with the large tank.

There are many considerations which dictate the use of a vertical drive, principally the simplicity with which the whole assembly can be mounted and handled. With the use of special jigs and fixtures, the assembling and dismantling of the test assembly are easily accomplished by one skilled mechanic.

The driving turbine was machined from a solid forging and is mounted on one air-oil-mist lubricated ball bearing, indicated as the main bearing in the sectional view. The blade sections were milled into the outer periphery of the disk, with nozzle passages arranged around the disk and drilled in to the inlet casing. The turbine rotor was drilled to receive a $^7/_{16}$-in. drill rod for driving the test disk. The drive shaft is attached to the turbine rotor by a simple pin arrangement near the top of the assembly, making it very easy to assemble the rotating parts. The drive shaft is attached to the test disk, and this assembly is put in place by sliding the drive shaft through the damper bearing and turbine rotor, then inserting the pin.

With the rotating elements in place, the lower half of the electric furnace is lifted into place and secured. The complete assembly is then ready for lowering into the test pit. A very good vacuum seal is of course necessary where the drive shaft passes through the vacuum cover. It was also found necessary to have something of a damper bearing on the otherwise one-bearing system to avoid large-amplitude vibrations at the first and very low critical speed. These two functions were combined. The damping arrangement was accomplished by placing the flange of the bearing shell on a flat surface and forcing these contact surfaces together with a nominal load applied with several compression springs. This allows the shaft and bearing to move if the vibration force is large enough, thus dissipating the energy in friction between these surfaces. The vacuum seal is accomplished simply by using a very close-fitting babbitt-lined bearing on the journal. This complete assembly is flooded with oil to prevent air leakage into the pit, and the oil which leaks through is thrown into a small reservoir around the shaft by an oil slinger, and piped away to the bottom of the pit to a water-cooled oil tank. It was found necessary to keep the oil cold to prevent vaporization at the very low pressures existing within the vacuum chamber.

The drive turbine was designed to operate on either 250-psig steam exhausting to 5-psig back pressure or 100-psig shop air, exhausting to atmosphere. To date practically all operation has been on air because of the greater convenience, but as test assemblies become too large, it is possible to revert to the steam power to obtain the greater driving power necessary.

The turbine element itself was designed to operate up to 60,000 rpm. It is protected by an electrical overspeed trip as part of the tachometer system (Standard Electric Time chronotachometer), and by a mechanical overspeed trip in case of a tachometer-generator failure. The electric trip is easily adjusted for varying operating conditions.

The test disk suspended from the drive shaft is enclosed in a refractory-lined electric furnace. The resistance-type heating elements are located above and below the path of rotation of the turbine blades. Each heating element is supplied with direct-current power from a "Flexarc" welding set, which affords excellent control of the power distribution. Brown Instrument Radiamatic radiation pyrometers are used to detect the disk temperature. The temperature gradient of the disk is measured by using four pickups focused at four radial positions on the disk, one on the rim, one on the center of the hub, and the other two on the profiled section of the disk.

For testing where large temperature gradients are desired, water-cooled coils of copper tubing are placed in a plane just above and just below the body of the disk. Each turn in the cooling coil is controlled separately to give complete flexibility of control of the temperature gradient.

In order that there be a uniformity of the emissivity of the test disks when using the radiation pyrometers for temperature detection, the disks are painted with Thurmalox No. 7 and heated to 1200 F in a temperature-controlled furnace to drive volatile matter from the paint. The radiation pyrometers are calibrated by making a complete test assembly, and, in addition, placing several thermocouples against the surface of the disks and calibrating with the disk stationary. The vacuum chamber is evacuated to operating pressure in order to include the effect of a low-density atmosphere on the pickups. The radiation pickups are water-cooled to isolate them from the high ambient temperatures within the pit.

Babcock and Wilcox Kaocast castable refractory is used for the furnace lining. The exposed surface of this material is painted with Johns Manville's Hellite to prevent dusting which upsets the pyrometer readings. It is of interest to note that these furnace elements must be cured for several hours with a temperature of 1200 F, in order to drive off all of the water in the refractory. Failure to do this results in loss of vacuum because of large quantities of water which accumulate in the pit during such a run.

The working space available in the pit (inside the vacuum wall shown) is 45 in. diameter and 33 in. deep. If a high-temperature test is to be run, the available working space is reduced by the space necessary for the furnace. The over-all depth can be increased to 6 ft by changing the vacuum tank.

A Kinney vacuum pump is used to evacuate the pit. The seal oil is supplied by a small gear pump, and a small amount of cooling water is supplied from the laboratory water system.

The temperature of the disk is controlled manually by adjusting the current setting of the welders and noting the temperature of the disk, the temperature of the heating-element nichrome ribbon, and the current flow to the heating elements. These quantities are indicated on instruments conveniently located to give the operator sufficient guidance to control the disk temperature properly. A Brown "Electronik" strip-chart recorder is used to indicate the disk temperature, and a Leeds & Northrup "Micromax" strip-chart recorder is used to indicate the heating-element temperature.

In addition to controlling the disk temperature and maintaining the lowest possible absolute pressure in the pit, the only other variable needing control is the disk speed. This is accomplished by taking an electrical signal from the Standard Electric Time tachometer circuit and feeding it into a Brown Electronik Air-o-Line recording controller. This in turn controls the position of an air-actuated throttle valve in the turbine-inlet line. Failure of either air or electrical power closes this valve.

These controls, along with several small flowmeters, temperature indicators, and valves to control the water flow to cooling coils, a manometer to indicate pit vacuum, and a tachometer to measure speed accurately are mounted on a panel in an adjoining room to the test pit. The test pit can be viewed clearly through a tempered plate-glass window which reduces the noise level in the operating room.

In order to eliminate the teardown and assembly time after each run, usually 5 hr, a long-focal-length microscope, Fig. 1, was purchased from Bausch and Lomb and is being placed in service. This will make it possible to record disk growth while the disk is running.

Test Procedure

For the purpose of evaluating many different types of materials in the shortest possible time, it was decided to standardize on the conditions of time and speed. Inasmuch as the military

running speed of the jet engine using this disk configuration is 12,500 rpm, it was decided to start the test at 12,000 rpm and increase the speed 500 rpm at each time interval of 5 hr. In this way the relative merits of each different disk material could be judged on an equal basis. This rotational speed results in stresses in the disk as indicated in Fig. 3. Jet-engine operating conditions were duplicated as closely as possible by using an actual turbine disk 13 in. diam, approximately 1 in. thick, and with a 4-in. bolt circle, and temperature gradients similar to those encountered in engine operation. These were established as 1200 F at the periphery and 1050 F at the center, Fig. 4.

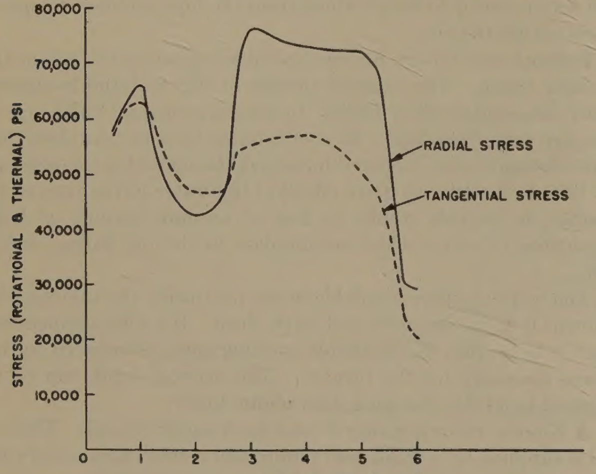

Fig. 3 Stresses, Including 150-Deg F Temperature Gradient, in Bladed Turbine Disk Rotating at 16,500 Rpm

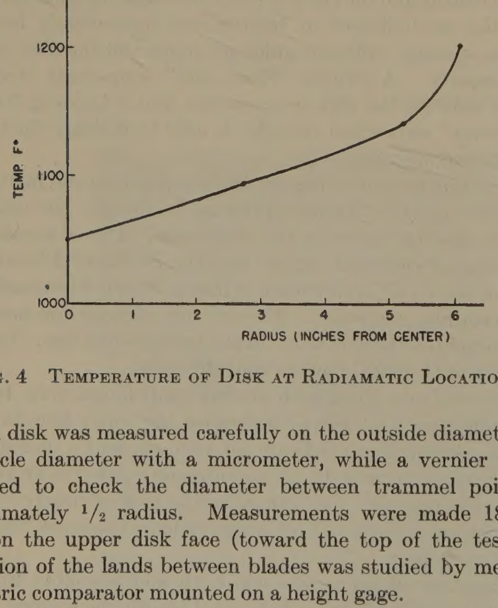

Fig. 4 Temperature of Disk at Radiamatic Locations

Each disk was measured carefully on the outside diameter and bolt-circle diameter with a micrometer, while a vernier caliper was used to check the diameter between trammel points at approximately $^1/_2$ radius. Measurements were made 180 deg apart on the upper disk face (toward the top of the test pit). Distortion of the lands between blades was studied by means of an electric comparator mounted on a height gage.

Before each new-type disk was run, a correlation was obtained between the disk temperature as measured by thermocouples placed against the stationary disk and the radiation pyrometers. The test was begun by heating the disk to the desired temperature while rotating at a low speed of 4000 rpm. This took about $1^1/_2$ hr after which the disk speed was increased rapidly to the desired level and held there for 5 hr. The disk was allowed to cool to room temperature before making any measurements.

Cast turbine rotating blades were taken from the same batch as those used for production engines and inserted in the grooves of the disk.

The first material selected for testing was quite naturally the standard material for aviation gas turbines built by the author's company, i.e., 19-9-DL conforming to AMS-5521. This is a hot-cold worked material, stress-relieved at 1200 F, so that the final hardness of the disk is 229-255 Bhn. It passed both an x-ray and reflectoscope test. The disk was forged by the Wyman Gordon Company.

The second disk tested was also a 19-9-DL material hot-cold worked and stress-relieved at 1200 F, but in this case the Brinell hardness was 285-311, which is considerably above the range of 228-269 required by AMS-5521. This disk was forged by the Midvale Steel Company to our instructions to develop a high strength and low hot ductility.

The third disk was also a standard material but not in engines of the author's company. It was a 16-25-6 Timken alloy, solution-treated and cold-forged to a hardness range of 240-300 Bhn by Canton Forge Company. This particular hardness was standard practice at the time these tests were started but current practice is somewhat lower. Brinell range of the disk being tested was 262-293, except for one spot at the center of the disk which was 217.

The fourth disk selected for testing was the material widely used in Great Britain, G18B. This material was hot-cold-worked to a Brinell hardness of 235-255 and was forged by the Crucible Steel Company. The processing of this disk was not in line with current practice, but an early attempt to duplicate the British material.

The chemical analysis and physical properties of the foregoing materials are listed in Tables 1 and 2.

Test Results

Table 3 shows the growth, speed, and time of operation of each disk. In Fig. 5 the growth of the outside diameter of the disks is plotted against time. All disks were spun at increasing speeds (500 rpm faster each 5 hr) until failure occurred. Exceptions were made of the standard 19-9-DL disk and G18B, where, because of fairly rapid growth toward the end of the test, the speed was held constant and the time increased.

Standard 19-9-DL. Failure occurred after a total time of 54.7 hr, and after a speed of 15,500 rpm had been reached. A total growth of the outside diameter amounting to 0.122 in. had occurred before a turbine blade pulled out of the disk. Figs. 6, 7, and 8 show the mechanism of failure through plastic flow of the disk lands as well as the condition of the disk at failure. This metal flow allowed the blade to slip out of position, and unbalanced the disk, which sheared the drive shaft. It is to be noted that not only had the blade slipped out of the disk lands but simultaneously many small cracks were appearing elsewhere in the disk. Table 3 also shows that many blades were becoming quite loose prior to failure.

Warning of impending failure was obtained in the following ways:

1 Rapid and large increase of the outside diameter of the disk.

2 Blade looseness.

3 Small cracks at various locations on the disk.

The ductility of this disk even after testing was quite high and showed room-temperature elongations of 23-29 per cent and a high-temperature elongation of 26 per cent.

Cold-Worked 19-9-DL. This disk failed as shown in Figs. 9 and 10, after a total time of 27 hr and after reaching a speed of only 14,000 rpm. The total disk growth was only 0.011 in. This failure was a very brittle break in which titanium nitrides were

TABLE 1 CHEMICAL COMPOSITION, HARDNESS, AND MILL HISTORY

	Chemical composition											Brinell Hardness		
Material	C	Mn	Si	Fe	Ni	Cr	Mo	Ti	W	Co	Cb	Rim	1/2 Radius	Center
19-9-DL Standard	.30	1.07	.60	Bal	8.30	18.86	1.34	.39	2.01		.46	255 255	235 241	229 235
19-9-DL Cold Worked	.30	1.09	.68	Bal	9.32	19.30	1.08	.43	1.54		.49	300 311	285 293	293 293
Timken 16-25-6	.09	1.72	.37	Bal	24.10	16.48	6.82					269 293	269 285	*217 262
G-18-B	.39	1.08	.82	Bal	14.80	14.88	2.80		3.01	11.42	2.98	235 241	241 255	241

Mill History	
19-9-DL Standard	Hot cold worked at temperature above 1350°F. Stress relieved at 1200°F.
19-9-DL Cold Worked	Hot cold worked at temperature above 1350°F. Stress relieved at 1200°F.
Timken 16-25-6	Solution treated and cold forged.
G-18-B	Upset forged to thickness 15% oversize. Reheat to 1300°F. for final hot-cold working to size. Stress relieved 1200°F., 6 hours air cooled.

* One spot at dead center—217.

TABLE 2 PHYSICAL PROPERTIES OF TURBINE DISKS AT ROOM AND ELEVATED TEMPERATURE

		Tensile Strength	Yield Strength .2% offset	Elong. in 2"	Reduction of Area	Stress for Creep to Rupture at 50 hrs.; lb/sq.in.; 1200°F.	Elongation at Rupture
Std. 19-9-DL Production Disc of Same Hardness as Test Disc		130,000	77,000	29%	38%	41,000	26%
19-9-DL Disc after Testing	Tangential	114,250	81,000	22.8%	24.1		
	"	114,750	80,750	27.4	25.8		
	Radial	116,500	82,500	28.2	20.6		
	-	115,750	80,500	28.8	33.1		
Cold Worked 19-9-DL Disc after Testing	Tangential	149,000	112,500	17.6%	25.7	50,000	2.25%
	"	150,500	112,500	16.4	24.8		
	Radial	150,500	113,750	16.5	25.4		
	"	148,250	112,500	15.6	24.4		
Timken 16-25-6 Disc after Testing	Tangential	131,000	98,200	9.8	10.8		
	"	130,000	99,600	11.2	16.0		
	Radial	129,000	96,800	9.6	14.6		
	"	130,000	96,400	10.4	14.6		
Timken 16-25-6 Disc similar to disc tested Specimen 1" from center		121,000	99,250	3.6	7.0	50,000	3%
G-18-B Disc Similar to Disc Tested	Tangential 1" from center	122,000	98,000	15	21		
		106,000	89,000	5	8	55,000	11%

found at the point of failure. These could have acted as points of stress concentration which in a brittle material such as this could have promoted failure.

It is to be noted that failure was sudden and without any of the warnings provided by the standard disk. This disk had a room-temperature elongation of 15-17 per cent in radial specimens taken after completion of the test, but only 2.25 per cent elongation in high-temperature tests, as shown in Table 2.

16-25-6 Timken Disk. After 50 hr and after reaching a speed of 16,500 rpm, this disk disintegrated. With a total growth of only 0.020 in. this disk literally "burst" into many pieces as shown in Fig. 11. The disk had a room-temperature elongation of only 9 per cent in radial test specimens taken after completion of the tests but only 3 per cent in high-temperature tests. The blocks shown in Fig. 11 were used instead of blades to apply load to the disk because of premature failure of the blades over 15,500 rpm. The stress on the disk was the same with the blocks as with the blades.

G-18-B Disk. After 37.5 hr operation and a total growth of 0.023 in. the test was stopped at a speed of 15,000 rpm. Because rapid growth was taking place, an additional run of 5 hr at the same speed was made. This caused the growth to continue at a rapid rate to 0.033 in. at which time the test was halted in order to prevent a wreck. The disk was showing the same rate of growth as the standard 19-9-DL disk but doing so at a slightly lower speed.

TABLE 3 DISK GROWTH AND BLADE LOOSENESS

SPEED RPM	19-9-DL Standard Time	Growth OD 1/2R Land	No. of Loose Blades	19-9-DL Cold Worked Time	Growth OD 1/2R Land	No. of Loose Blades	Timken 16-25-6 Time	Growth OD 1/2R Land	No. of Loose Blades	G-18-B Time	Growth OD 1/2R Land	No. of Loose Blades
12,000	5	.000 .000 .001	0	5	.003 .001 .002	2	5	No		5	No	4
12,500	5	.000 .000 .002	0	5	.005 .002 .003	2	5	Measurements		5	Measurements	4
13,000	5	.000 .001 .002	0	5	.009 .002 .005	2	5	"		5	"	4
13,500	5	.001 .000 .002	0	5	.011 .002 .009	2	5	.007 .000 .010		5	.004 .002 .011	4
14,000	5	.003 .001 .004	0	3.2 (1)	.011 .004 .012	2	5	.008 .000 .012		5	.006 .002 .015	4
14,500	7.5	.008 .003 .007	2				7.5	.010 .000 .014	1	7.5	.011 .005 .023	8
15,000	5	.016 .010 .015	9				5	.013 .001 .017	3	5	.023 .015 .043	8
	5	.018 .012 .020	10							5 (2)	.033 .023 .058	
15,500	5	.036 .024 .038	25				5	.015 .002 .024	7			
	5	.085 .043 .087	34									
	2.2 (1)	.121 .077										
16,000							5	.023 .010 .036	7			
16,500							2.5 (1)					

(1) Disk failed.
(2) Test stopped; rapid creep.

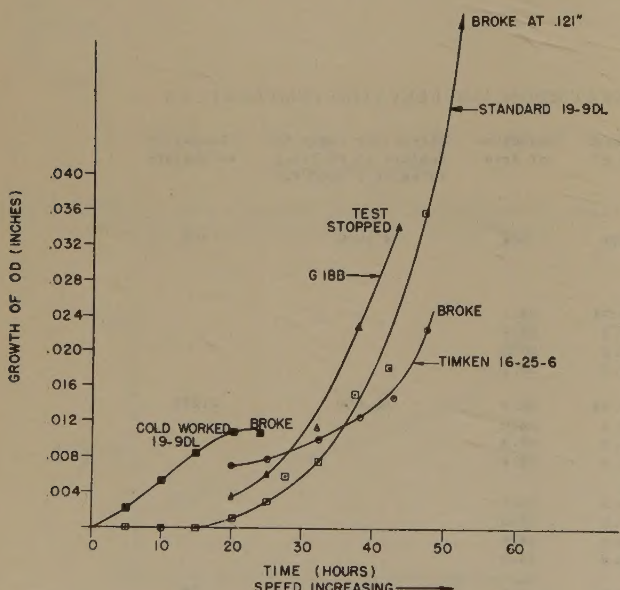

Fig. 5 Comparative Growths of Turbine Disks

Fig. 6 View Showing Plastic Flow of Lands for Standard 19-9-DL Disk

Discussion

In examining the types of failures in the several disks, it was observed that the standard 19-9-DL disk, Fig. 6, showed extensive plastic deformation of the small lands holding the blade in place. With increased movement of the lands the blade exerted increasingly higher unit pressure on the land and eventually pulled out of the disk. Cracking also occurred throughout the rest of the disk at many locations. Of perhaps greatest significance are the cracks shown in Fig. 7 where a large crack caused by radial stress has been circled at $2^5/_8$ in. radius. It was noted that this corresponded to the point of highest radial stress as indicated in Fig. 3. Cracks caused by tangential stress were also seen within the bolt circle and near the boltholes. From Fig. 3 this area is shown to be the one with highest tangential stress.

Failure of the cold-worked 19-9-DL disk, Figs. 9 and 10, must be attributed to one of three reasons: (*a*) Titanium nitrides at the point of failure; (*b*) unequal loading of the disk lands while driving the blade into the disk; or (*c*) low ductility, coupled with either reason (*a*) or (*b*). The break was certainly not a ductile failure. Additional testing is indicated for this material in the same hardness range and for the purpose of substantiating this test.

The failure of the Timken disk, Fig. 11, showed: (*a*) that from the center of the disk to the boltholes the break was normal to the plane of the disk, indicating a very brittle fracture (4); (*b*

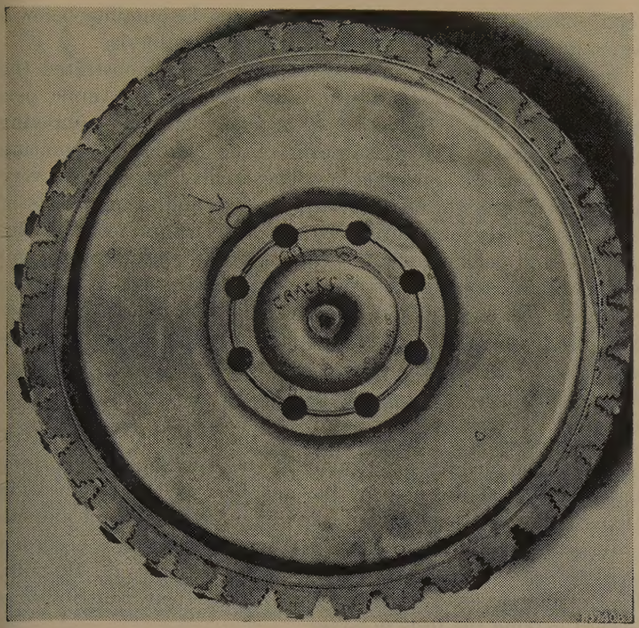

FIG. 7 CONDITION OF DISK AT FAILURE OF STANDARD 19-9-DL MATERIAL

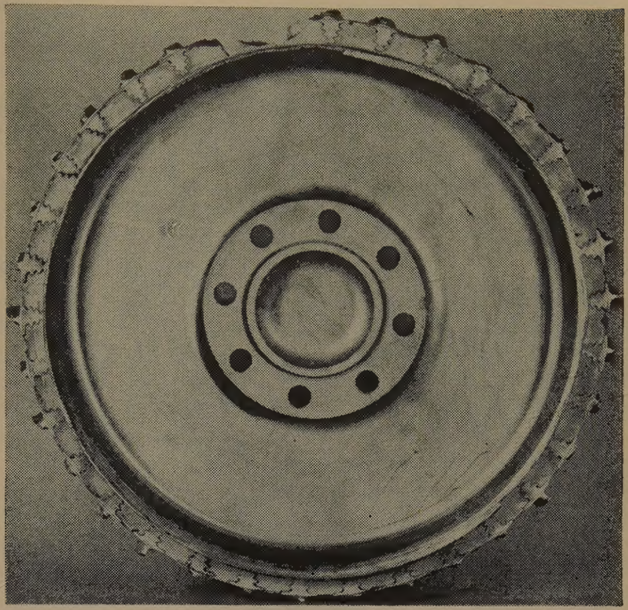

FIG. 9 COLD-WORKED 19-9-DL MATERIAL DISK FAILURE

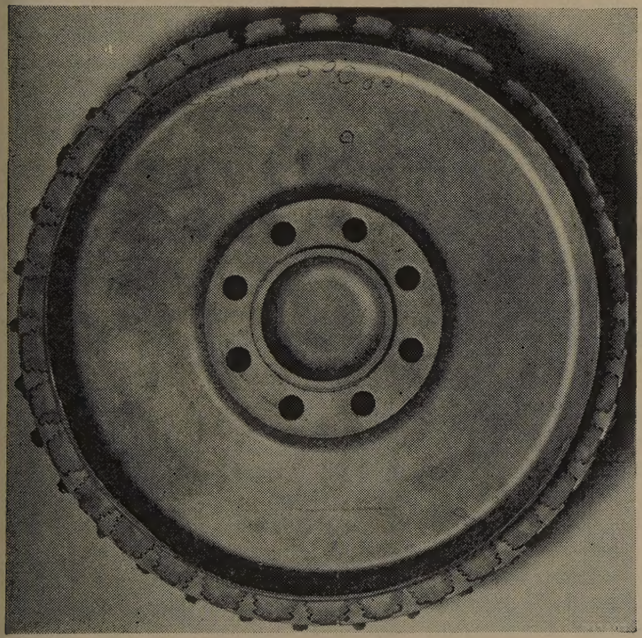

FIG. 8 CRACKING OF DISK NEAR RIM IN STANDARD 19-9-DL TESTS

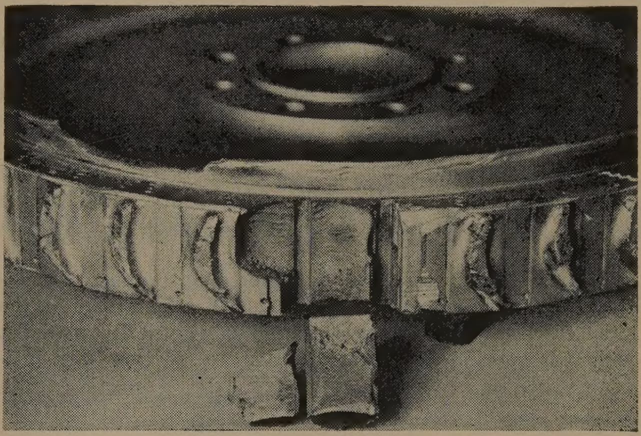

FIG. 10 COLD-WORKED 19-9-DL DISK FAILURE SHOWING FRACTURE IN BLADE ROOT FASTENING

from the boltholes out to the periphery the fracture tended to be more of a "cup-and-cone" break indicative of some ductility. The whole break appeared to be a failure caused by tangential stress and originating about the boltholes. It is recognized that a crack caused by radial stress could start just outside the bolt circle and then change direction by the action of the tangential stress.

CONCLUSIONS

1 The disks herein reported represent the first of a series in which many of the best high-temperature disk alloys are to be tested under the same accelerated test schedule. From this program it is expected to establish the relative merits of the various available disk materials.

2 The most promising material or materials determined by the program so far can then be tested under constant-speed conditions to obtain data which can be correlated with existing creep-rupture information.

3 This program procedure will give reliable guidance to the selection of the "strength versus ductility" relationship which represents the best compromise for each application of a specific disk material.

4 The characteristic type of failure appears to have a direct relationship to the magnitude of the ductility for a given material. Further experience may indicate sufficient reason for specifying a minimum ductility for a given disk material for a given engine application.

5 A ductile disk will give ample warning of impending failure by means of blade looseness, large and rapid growth of the diameter, and cracking at various locations over the face of the disk. This means that if a bladed turbine disk were operating in a

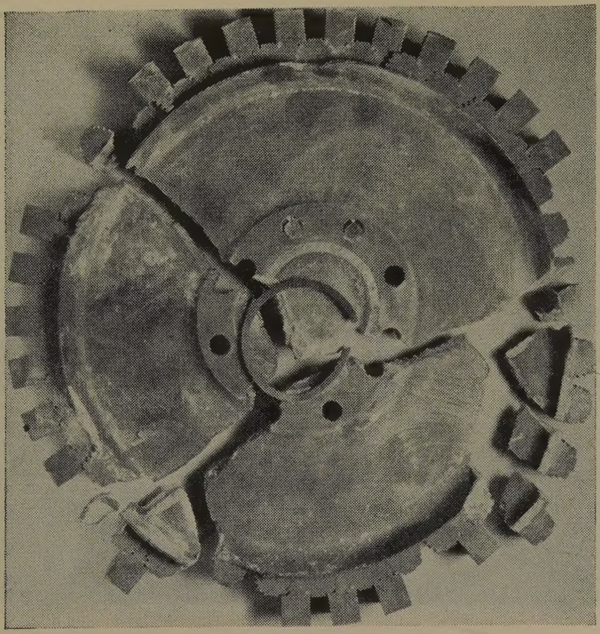

Fig. 11 Disk Burst in Test of 16-25-6 Timken Alloy After 50 Hr, and Speed of 16,500 Rpm Being Reached

flying engine, the growth would be most readily detected by a blade-tip rub, thereby giving warning prior to a failure.

6 A brittle disk will give little or no warning of impending failure. Additional tests must be run before a minimum ductility range can be established.

7 Of the four disks reported, the standard, 19-9-DL showed the best combination of strength and ductility, although the Timken 16-25-6 material as processed was slightly stronger than the standard 19-9-DL.

8 These tests emphasize that processing (forging or heat-treatment) can be specified to produce a wide range in the strength versus ductility relationship in any given disk material, and that metallurgists should strive to develop disk materials of greater strength without reducing the ductility below a safe minimum which will avoid brittle breaks in service.

9 The operation of the disk test pit has demonstrated the practicability of hot-spin-testing turbine rotors. Ample personnel safety is relatively easy to provide while not hampering the test program. The measurement of the test-disk temperature is straightforward and reliable. The facility has proved very adaptable to tests of disks of various sizes and shapes, tests of bolting methods, and relatively low-temperature tests of aluminum- and magnesium-alloy disks. Since disassembly and reassembly time is relatively short, creep-rupture data on turbine rotors can be accumulated rather rapidly and at relatively low cost. Damage as a result of disk failure has been less than anticipated.

Acknowledgments

The authors wish to acknowledge the valuable help contributed by the operating personnel at the Aviation Gas Turbine Laboratory, and the help of the members of the Engineering Department for their part in the design of the test pit and coordination of the tests run. Special commendation is accorded to Messrs. W. J. Staton, J. Markell, P. Burgess, H. Hanzlik, P. J. Gillespie, and William Durkin for their contributions toward making the tests possible.

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Comparison of High-Temperature Alloys Tested as Blades in a Type B Turbosupercharger

By W. C. Stewart[1] and H. C. Ellinghausen,[2] Annapolis, Md.

The authors describe briefly a series of jet tests, utilizing gas produced from the combustion of Diesel fuel oil, as a means for comparing the resistance of a number of high-temperature alloys to hot-gas impingement. The deficiencies of this method for simulating conditions in a gas turbine are discussed. The paper is concerned principally with comparative tests of a number of high-temperature alloys when tested in the form of blades in a Type B turbosupercharger. The test rotor contains 142 blades, representing 12 different alloys. Both wrought and precision-cast blades are included. Tests were made at eight temperatures, ranging from approximately 1200 to 1500 F, test runs at each temperature being of 50 to 150 hr duration, except in the case of the 1500 F test run, which was continued for 1000 hr. After several of the test runs, measurements were made to determine the amount of permanent extension in the blades and disk. The extension of the blades and disk accompanying progressively higher testing temperatures is shown graphically. A procedure for correlating the metal temperature of the blades with that of the combustion gas in the nozzle chamber of the supercharger is described.

Introduction

During the period 1940–1947, a great deal of effort and a large amount of money were expended in the development and improvement of a class of high-temperature alloys now sometimes referred to as "superalloys." These alloys were developed primarily for turbosuperchargers, gas turbines, and jet engines. For these applications, strength and stability at elevated temperatures are considered of first importance. Continued efforts were made to enhance the high-temperature strength of these materials. While this was being accomplished, designers were equally diligent in their determination to design for higher operating temperatures and stresses, in order to increase efficiency of operation and reduce weight to a minimum.

In this development program, a great many individual alloys were made, and many thousands of tests were conducted on forged, cast, and precision-cast alloys. Governmental agencies, industrial concerns, educational institutions, and research laboratories co-operated in this work in a most praiseworthy manner. Test data obtained through this co-operative effort are summarized in a large number of reports. Some of these reports were issued by the Office of Scientific Research and Development, some by the Bureau of Ships, and Bureau of Aeronautics, and others by private companies. Recently, the classification status for some of these reports was modified, and large portions have appeared as technical publications.

The interest of the Bureau of Ships in these high-temperature alloys rests mainly on possible applications for gas turbines. Consequently, test programs sponsored by the Bureau of Ships were directed along these lines. The construction of gas turbines for ship propulsion presents quite different metallurgical and engineering problems from those encountered in the production of superchargers and jet engines. Gas turbines for ships have to be designed for much longer operating life with a minimum of upkeep. Ship machinery must be designed and constructed to resist shock occasioned by gunfire, as well as explosions resulting from bombs and depth charges. Parts for large gas-turbine units, such as the rotor and casing, are of much heavier section than are similar parts for superchargers and jet engines. These circumstances have introduced special metallurgical and engineering problems in the development program of the Bureau of Ships.

Most of the high-temperature test results reported by the co-operating activities were obtained with specimens prepared from small-diameter bars, small castings and forgings, and specimens precision-cast to size. On the other hand, an important part of the program of the Bureau of Ships consisted of fabricating relatively heavy forgings of several of the high-temperature alloys, in order that these might be sectioned and tested, and results compared with those obtained for bar stock and small sections. Likewise, it was necessary to ascertain whether heat-treatments, which had proved effective for light sections, were applicable in the case of heavy sections. The welding of large disks presented new problems which required investigation. The feasibility of casting rotor disks of high-temperature alloys was investigated. Later, the work was extended to include the casting of disks with integral blades. At the present time ways for forming hollow blades are being studied.

Practically all of the creep and stress-rupture data reported for the high-temperature alloys were obtained in test furnaces containing air. Early in the investigation it was proposed that similar tests be made in a gas atmosphere similar to that produced by combustion in a gas turbine. The practicability of adapting the usual types of creep and stress-rupture test units for conducting tests in combustion-gas atmosphere was considered. This examination indicated that it would be difficult to maintain the desired composition of gas, let alone simulate velocity conditions encountered in a gas turbine. It was decided that these conditions could best be met by operating a turbosupercharger as a gas turbine. Aircraft turbosuperchargers were readily available, and steps were taken to adapt one for the purpose. It appeared that the value of such a test would be enhanced if a special rotor containing blades of a number of selected alloys was provided. In this way a direct comparison of several alloys in the form of blades would be obtained under identical test conditions.

The first experimental unit for simulating the impinging action

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Note: The opinions expressed in this paper are those of the authors and do not represent necessarily the opinions of the Navy Department nor of the Society. Paper No. 48—A-96.

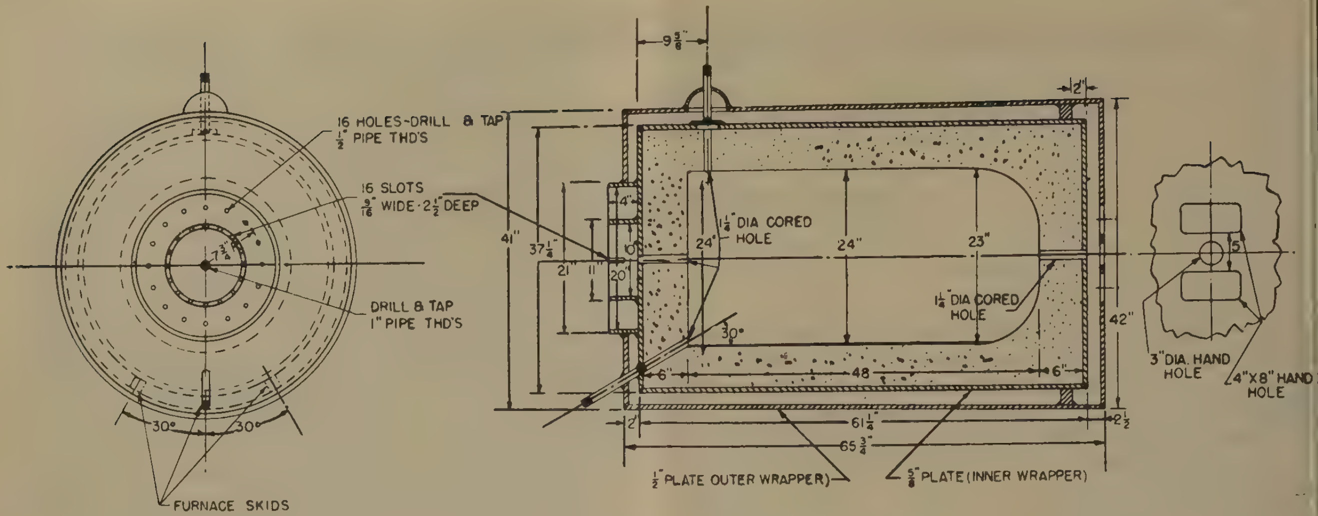

Fig. 1 Combustion Chamber for Jet Erosion Tests

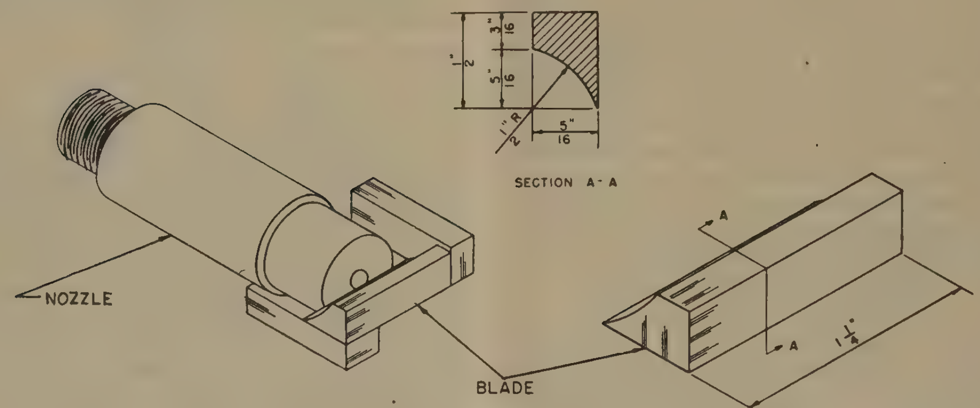

Fig. 2 Nozzle and Specimen Assembly for Gas Erosion Test

of high-velocity high-temperature gas on gas-turbine alloys is shown in Fig. 1. The assembly consists of a double-wall cylindrical combustion chamber, the inside of the interior shell being lined with refractory material. The shells are of carbon-steel plates, except for the head which is of 18:8 chromium-nickel steel. Fifteen nozzle assemblies, each consisting of a nozzle and fixture for holding the test specimen, are mounted radially on a common ring support which is attached to the exhaust end of the combustion chamber. Details of one of these nozzles and simulated blade section are shown in Fig. 2. Gas is produced from Diesel fuel oil, combustion taking place under 40 psi pressure. Oil is fed into the combustion chamber from a tank maintained under air pressure. The flow of oil is controlled by means of a pressure-operated needle valve actuated from a thermocouple which is mounted in a dummy nozzle. Air is supplied through suitable reducing valves.

Tests were made on a number of alloys both wrought and cast at 1300, 1350, 1400, 1500, 1600, and 1650 F temperatures. Eight test runs were made at these temperatures, the duration of test ranging from 155 to 1370 hr. The gas velocity at the jets for these tests ranged from 1300 to 1500 fps. Normally, the gas is directed against the apex of the blade so as to sweep upward as shown in Fig. 2. However, for one set of specimens, the gas was directed at a flat surface of the blade section as well as at the apex. Results of gas analysis for two of the test runs are given in Table 1.

The percentages of constituents in the gas remained quite uniform throughout the tests. For test run No. 8 the carbon-dioxide content of the gas was reduced to about $3^1/_2$ per cent, and the oxygen increased to about 14 per cent, in order to increase the oxidizing effect of the gas.

TABLE 1 GAS ANALYSIS, PER CENT

Test run	CO_2	O_2	N_2	CO	H_2	Methane group	S (calculated as SO_2)	Number of determinations
1	5.2	12.3	bal.	.3	.4	.15	0.051	4
6	6.2	12.6	bal.	..	..	...	....	7

Of some thirty-six alloys tested in this series, only one revealed any serious damage. In that case, test conditions of 1350 F gas temperature and 1380 fps gas velocity caused the test section to swell up and exfoliate at the edges. A similar phenomenon has been observed in connection with another alloy containing a substantial percentage of molybdenum, but under less severe test conditions. In this instance, disintegration of the alloy occurred during a scaling test carried out in the muffle of an electrical-resistance furnace containing air. It should be mentioned that no similar deterioration of these two alloys has been encountered in test units operated as gas turbines.

During one test run, the temperature-control mechanism failed to function, with the result that the impinged areas of the test

sections attained a temperature of approximately 2000 F for about 20 min. This accidental sojourn at excessively high temperature caused V-shaped notches to be cut in the blade sections in the path of the impinging jet. All materials were affected similarly, with the result that no information was obtained as to the relative resistance of the several alloys to hot-gas impingement.

At the beginning of the test it was feared that the test sections might be eroded as a result of small particles becoming dislodged from the ceramic lining of the combustion chamber and being carried along by the gas stream. Inspection of the edges of the simulated blade sections in the early tests revealed some evidence of erosion from these particles, but, with continued operation, trouble from this cause apparently diminished. Should the jet test have indicated important differences in erosion resistance for the several alloys, the role played by nonmetallic particles would have been cause for considerable apprehension. However, it was concluded that in this type of test equipment, effort should be made to eliminate the usual ceramic lining materials from the combustion chamber.

Tests of Blade Materials in an Aircraft-Type Turbosupercharger

Experience with the jet tests emphasized the importance of testing gas-turbine alloys in hot-gas environment while under stress. The adaptation of available creep and stress-rupture test units for operation in a combustion-gas atmosphere was considered impracticable. One of the perplexing problems was the chemical composition of a gas for simulating service conditions, and the maintenance of a constant supply of the gas over a long period of time. Even with the solution of this problem, no information as to the effect of high-velocity gas on metal under stress would be obtained. For these reasons the proposal to conduct creep and stress-rupture tests in combustion gas was abandoned. However, it appeared feasible to conduct such tests *in vacuo* and in oxygen.

The next step was to install a Type B turbosupercharger in conjunction with a specially constructed combustion chamber for operation as a gas turbine. The supercharger rotor provided for test contained 142 blades representing twelve different alloys. The rotor disk is of forged 16:25:6 nickel-chromium-molybdenum (Timken alloy). The supercharger is mounted on a stand bottom-side up from the way it is installed in an airplane. The combustion chamber is of the double-wall type, approximately 15 ft long × 24 in. OD, made entirely of molybdenum-bearing 18:8 chromium-nickel steel by welding. Air for combustion is supplied from an independent blower, and is introduced into the outer space of the chamber near the forward end. Air on entering the outer space flows to the rear where combustion takes place. The combustion gas passes through the inner chamber and enters the nozzle box of the supercharger. This arrangement permits preheating of the air, and at the same time reduces the metal temperature of the inner wall of the chamber. The unit is fired with Diesel fuel oil conforming to Navy Department Specification 7-0-2d, dated October 1, 1943. For purposes of safety the test unit is installed in a concrete-lined pit several feet below ground level.

In operating the supercharger, the oil-supply tank is maintained under 10 psi air pressure so that oil enters a rotary gear pump at this pressure. Oil leaves the pump at 300 psi pressure, but a sufficient volume is by-passed so as to reduce the pressure to the desired value. For 1500 F operation, the oil pressure is maintained at approximately 185 psi. Temperature control is accomplished by regulating a by-pass. The combustion chamber is fired by means of a Babcock and Wilcox "Mayflower" pressure fuel-oil-atomizing nozzle and sprayer plate.

Two thermocouples are mounted in the nozzle box of the supercharger. These are placed diametrically opposite, one on the small side and the other on the large side of the box. Thermocouples installed at these locations, at the present time, consist of No. 14 gage chromel-alumel wire contained in porcelain insulators assembled in an 18:8 chromium-nickel-steel tube. The hot junction of the element is exposed by means of slots milled in the protective tube. The hot-junction bead is encased in platinum sheet metal formed by collapsing a platinum tube around the element. This platinum sheath, in conjunction with a chromel wire connecting onto the end of the tube, also serves as a support for the element. This thermocouple represents one of the more advanced designs produced by the National Bureau of Standards. Temperature conditions for test are established by means of these two thermocouples.

A third thermocouple, completely enclosed in a 25:20 chromium-nickel-steel tube, is located in the gas stream ahead of the supercharger intake flange. This thermocouple serves as a control for operating the unit. Being completely enclosed, it is not subjected to conditions which cause deterioration and failure in thermocouples which are exposed to high-temperature high-velocity gas. The unit is provided with several other thermocouples but these are not required for operation of the equipment. In addition to the thermocouples, a wide-angle radiamatic pyrometer is installed in the stack above the supercharger, and focused on the bladed section of the rotor. The length of the major axis of the target area is 1.2 in. covering practically the full width of the blade band. A schematic diagram of the supercharger installation is shown in Fig. 3.

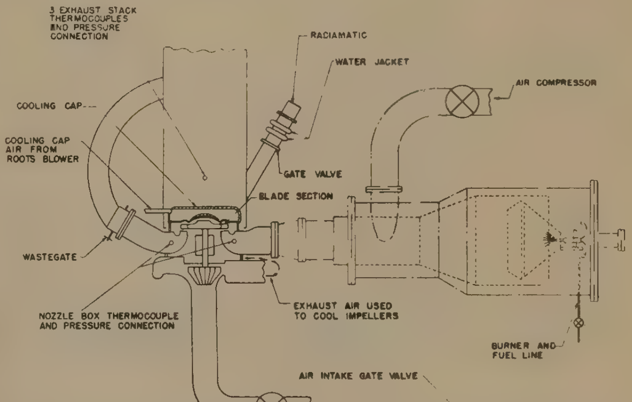

Fig. 3 Turbosupercharger and Combustion Chamber

The first specially constructed bladed rotor, designated No. 1, is shown in Fig. 4. Designations A, B, C, and E denote the location of scribed reference circles which serve as base lines for extension measurements. Each blade is numbered for identification, but it is usually necessary to renew these following each test run. The blades are held in the rotor mechanically, the blade roots being of the bulb type, one half with short necks, and the other half with long necks.

The chemical composition and tensile strength for the various blades of rotor No. 1 are given in Table 2.

Tests of rotor No. 1 were made at eight temperatures, ranging from approximately 1200 to 1500 F. Test runs at each temperature were of 50 to 200 hr duration. The supercharger was operated at an average wheel speed of 15,400 rpm for all test runs. This speed produces a maximum stress of 10,000 psi at a critical section of the blade due to centrifugal force. The only work performed by the unit other than to overcome friction is that re-

TABLE 2 CHEMICAL AND PHYSICAL PROPERTIES OF SUPERCHARGER BLADES

NAME OF ALLOY	CHEMICAL ANALYSIS												HEAT TREAT	TENSILE STRENGTH OF BUCKETS P.S.I.	
	C	Mn	Si	Cr	Ni	Mo	W	Ti	Cb	Co	N_2	Fe		ROOM TEMP.	1500 °F.
S-495 HT. 22792	0.41	0.53	0.47	13.87	19.89	4.40	3.54		3.51				2300°F 40 MIN. W.Q. 1475 °F. 16 HRS. A.C.	L.N. 152 200 148 950 S.N 132 300 146 500	L.N. 59600 65400 SN 63500 66900
N-155 LOW CARBON HT. 11534	0.15	1.68	0.31	21.50	18.99	2.90	2.08		1.06	17.60	0.06		SOL. TREAT 2240° 1/2 HR. A.C. AGED 50 HRS 1350 °F	L.N. 118 900 119 100 S.N. 115 100 117 900	L.N 66 500 70 300 S.N 64 900 65 100
N-155 HIGH CARBON HT. 11535	0.31	1.58	0.33	21.12	18.97	2.90	2.17		1.12	17.70	0.07		SAME AS LOW CARBON	L.N. 136 000 134 400 S.N. 134 100 134 000	L.N. 75 000 76 500 S.N. 75 650 74 400
19-9 DL B10658	0.29	0.89	0.54	19.34	8.51	1.28	2.10	0.22	0.11				50 HRS. 1350° F.	L.N. 132 900 127 600 S N. 127 000 126 000	L.N 54 600 53 250 S.N 56 800 54 000
K42B KB 476	0.05	1.16	0.36	16.78	42.68			2.70		21.23		15.04	2100 °F 1 HR. O.Q. 1500 °F. 20 HRS. A.C.	L.N. 139 400 135 000 S.N 125 000 127 800	L.N. 98000 90900 S.N 91400 89600
S-590 HT. 42883	0.43	0.69	0.09	17.80	19.64	3.82	3.94		2.75	18.37			2300 °F 40 MIN. W.Q 1500 °F 16 HRS. A.C.	L.N. 129 100 131 000 S.N. 133 800 135 400	L.N. 79 800 75 400 S.N 77 000 71 450
16-25-6 HT. 11542	0.08	1.79	0.58	14.98	25.40	6.00					0.15		24 HRS. 1300 °F.	L.N. 137 800 139 250 S.N. 143 000 135 900	L.N. 73 600 70 500 S.N 65 700 71 300
REFRACT-ALOY HT. M406	0.15	1.21	0.29	20.64	20.71	8.40	4.46			31.41		12.95	2300 °F. 4 HRS. O.Q. 1500 °F. 240 HRS. A.C.	L.N. 141 000 143 400 S.N. 123 700 115 200	L.N. 97 000 94 500 S.N. 89 400 95 800
CAST 61 ALLOY	0.43			24.66			5.18			67.73			USED AS CAST		72 400 69 600
CAST VITALLIUM	0.25			28.28		5.63				63.84			"		69 000 69 400
CAST 6059 ALLOY	0.39			24.61	33.15	5.34				33.05			"		60 000 63 500
CAST 422-19 ALLOY	0.51			27.56	15.51	6.59				47.61			"		62 000 62 000

NOTE: LN - LONG NECK BUCKET
SN - SHORT NECK BUCKET

quired to drive the impeller. The air furnished by the impeller is directed between the rotor casing and impeller casing to cool the bearings.

The volume of one side of the supercharger nozzle box is greater than the opposite side. Early in the investigation it was found that the gas temperature on opposite sides of the chamber differed somewhat. The average gas temperature was estimated arbitrarily by taking two thirds of the average gas temperature in the large side of the chamber plus one third of the average gas temperature in the small side of the chamber. The metal temperature of the blades was estimated to be 15 to 20 deg F lower than the average gas temperature. Procedures for determining blade-metal temperatures are described later in the paper.

Table 3 gives a summary of temperature conditions and length of test for the several test runs.

Data for a typical test run (No. 8) at 1500 F temperature are given in Table 4.

After each test run, the supercharger was removed from the test stand and taken into a constant-temperature room where measurements were made to determine permanent extension in the blades and rotor disk. Measurements were obtained on a Zeiss universal measuring machine, and were taken between

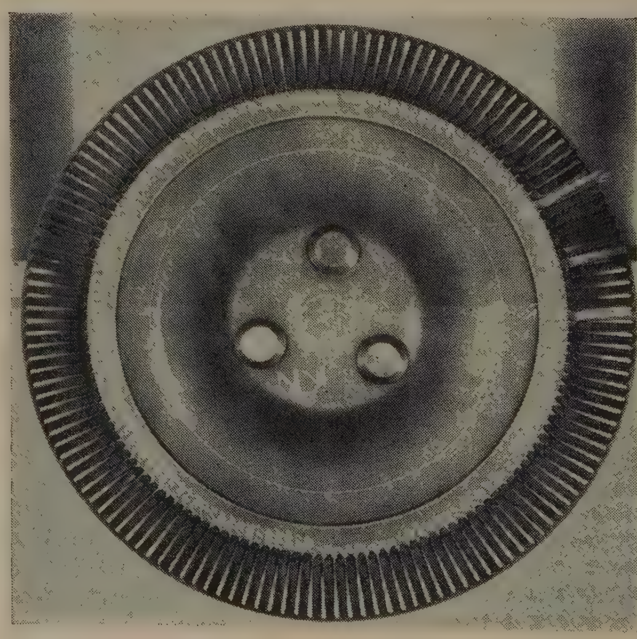

FIG. 4 TEST ROTOR NO. 1, CONTAINING BLADES OF TWELVE ALLOYS

reference circles E and A, A and B, and B and C. In measuring between circles E and A, each blade was measured, and the average taken. Two complete sets of measurements were obtained during the course of test run No. 6. A summary giving results of measurements for the various test runs is contained in Table 5.

The set of measurements designated run No. 6A was obtained following the first 100-hr test period at 1450 F temperature, whereas the set designated run No. 6B was obtained on completion of the 150-hr run. The average extension for each group of blades, obtained by averaging the readings for each blade in the group, are represented graphically in Fig. 5.

On completion of run No. 6, the 19-9 DL blades were removed from the rotor because of the considerable increase in creep rate following the second period of the run. It was feared that the 19-9 DL blades might fail if tested at higher temperature with the possibility of damaging the unit.

No new blades of high-strength alloys were available for replacement, and a set of vitallium blades was taken from a standard rotor for this purpose. This proved unfortunate as one of the replacement blades failed during the second 50-hr test run at 1500 F. It was known that the rotor from which the replacement blades were taken had been in service, but no other history was available. The failed blade segment damaged a considerable number of blades in the rotor. Forty-five of the damaged blades were removed and replaced with new vitallium blades. Fortunately, the damaged blades were fairly well distributed among the several alloys in the wheel so that despite the casualty each alloy was represented by at least six blades.

Renewal of forty-five blades and damage to the reference circles made it necessary to re-establish the reference marks, renew the bearings, and balance the rotor. Then the test was continued at 1500 F temperature with the view to obtaining 1000 hr of operation at this temperature.

Average extension values obtained for the blades following the several test runs at 1500 F, are also plotted in Fig. 5. It is observed that all of the temperature extension curves show a dip in the vicinity of 1300 F temperature. This behavior might be attributed to two causes: (a) Thermal adjustment in the rotor and blade roots may have resulted from the test conditions. Considering that extension measurements were taken between reference circle A on the disk and the tip of the blades, it will be seen that any such readjustment would be reflected in the measurements. (b) Precipitation occurs in many of these alloys when held for a considerable period at temperatures in the vicinity of

TABLE 3 SUMMARY OF TEST CONDITIONS

Test run	Gas-temperature conditions, deg F — Nominal	Average	Duration of test run, hr	Estimated blade-metal temperature deg F
1	1200	1196	100	1181
2	1300	1310	100	1295
3	1350	1347	50	1332
4	1400	1396	50	1381
5	1400	1411	150	1396
6	1450	1464	150	1449
7	1500	1510	100	1495
8	1500	1518	100	1503
9	1500	1514	100	1499
10	1500	1523	146	1508
11	1500	1519	200	1504
12	1500	1520	233	1505
13	1500	1517	123	1502

TABLE 4 TEST DATA FOR RUN NO. 8 AT 1500 F TEMPERATURE; DURATION OF RUN 100 HOURS

	Temperature, deg F — Maximum	Minimum	Average
Nozzle box—small side of chamber....	1530	1490	1505
Nozzle box—large side of chamber.....	1540	1500	1525
3/4-in. from exhaust edge of blades.....	1460	1420	1430
Temperature 6 in. from exhaust side of wheel — center of stack....	1325	1300	1315
1/2 radius of stack	1360	1330	1340
2 in. from wall...	1340	1315	1328
Air-duct pressure, avg in. Hg.........		10.3	
Stack pressure, avg in. Hg...........		0.4	
Pressure drop between nozzle box and stack, avg in. Hg................		9.1	
Nozzle-box pressure, avg in. Hg.......		9.5	
Speed of rotor, rpm.................		15550 max 15200 min 15450 avg	
Stress at critical section of blade, psi...		10200 max 9850 min 10000 avg	

TABLE 5 SUMMARY OF BLADE AND DISK MEASUREMENTS

NO. OF BLADES RUN 8	MATERIAL	TOTAL CHANGE IN LENGTH (INCH) — AFTER RUN 2	AFTER RUN 3	AFTER RUN 4	AFTER RUN 5	AFTER RUN 6A	AFTER RUN 6B	AFTER RUN 8	AFTER RUN 9	AFTER RUN 11	AFTER RUN 12	AFTER RUN 13
8	VITALLIUM	-.0012	.0000	+.0001	+.0007	+.0010	+.0012	+.0020	+.0025	+.0032	+.0041	+.0049
45	NEW VITALLIUM							+.0024	+.0031	+.0037	+.0057	+.0062
10	ALLOY 61	-.0011	.0000	-.0003	+.0005	+.0005	+.0008	+.0015	+.0019	+.0024	+.0027	+.0034
6	ALLOY 6059	-.0010	+.0001	-.0005	+.0005	+.0005	+.0010	+.0018	+.0022	+.0029	+.0040	+.0047
8	ALLOY 422-19	-.0008	+.0002	-.0003	+.0008	+.0007	+.0013	+.0019	+.0023	+.0027	+.0031	+.0038
9	REFRACTALOY	-.0007	+.0007	+.0003	+.0015	+.0014	+.0016	+.0021	+.0025	+.0029	+.0035	+.0042
8	ALLOY K-42-B	-.0014	.0000	-.0003	+.0003	+.0002	+.0007	+.0012	+.0015	+.0018	+.0020	+.0024
10	HIGH CARBON "N-155"	-.0008	+.0004	+.0002	+.0001	+.0010	+.0013	+.0018	+.0023	+.0027	+.0029	+.0038
8	ALLOY S-590	-.0009	+.0001	+.0001	+.0010	+.0013	+.0013	+.0018	+.0024	+.0028	+.0033	+.0040
8	ALLOY S-495	-.0003	+.0005	+.0004	+.0013	+.0014	+.0017	+.0026	+.0031	+.0037	+.0051	+.0056
10	LOW CARBON "N-155"	-.0007	.0000	-.0001	+.0007	+.0007	+.0011	+.0018	+.0020	+.0026	+.0031	+.0035
12	ALLOY 16-25-6	-.0005	+.0009	+.0007	+.0021	+.0021	+.0029	+.0039	+.0045	+.0052	+.0071	+.0075
NONE	ALLOY 19-9DL	-.0006	+.0007	+.0007	+.0023	+.0047	+.0080	REMOVED				
		INCREASE IN/IN.										
	DISTANCE BETWEEN "A&B"	.0014	.0014	.0013	.0012	.0017	.0012	.0015	.0014	.0014	.0014	.0014
	DISTANCE BETWEEN "B&C"	.0000	.00196	.00136	.00189	.00169	.00166	.00250	.00195	.00210	.00200	.00200

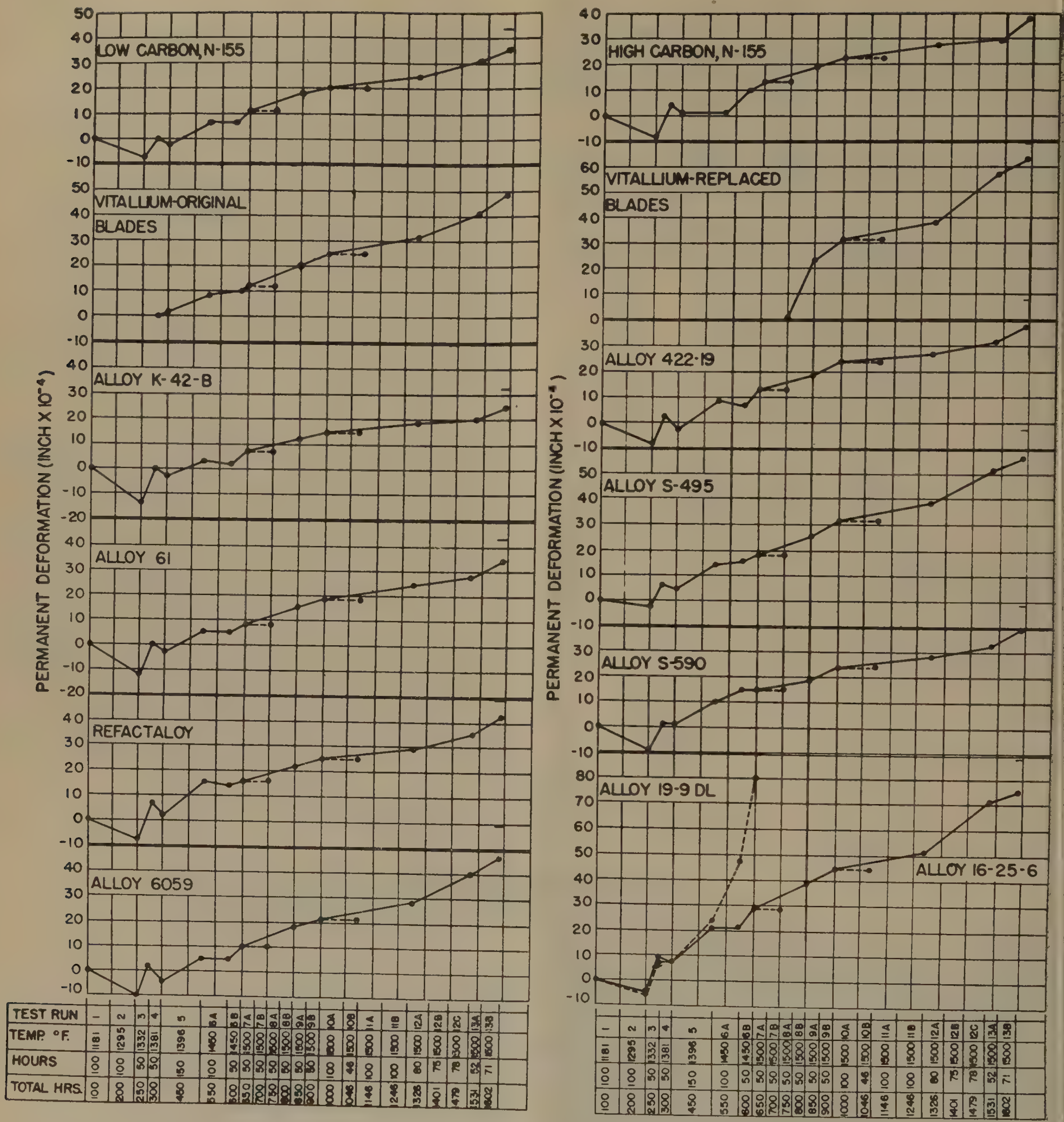

Fig. 5 Graphs Showing Extension of Turbine Blades at Various Testing Temperatures

1300 F, and dimensional changes might have accompanied this phenomenon. However, temperature-expansion curves obtained for a number of alloys of these types, have revealed no abrupt changes for relatively short times at temperature.

The short horizontal lines shown on the graphs for each alloy at run No. 11 represent the maximum extension which would have been obtained had the extension measurements for test runs Nos. 7 and 10 not been lost. As previously mentioned, readings for run No. 7 following a blade failure proved unreliable, as did readings for run No. 10 after a thermocouple tube end failed and rubbed on the blades. Extension rates for these two runs were obtained by taking the average of the rates for the preceding and following test runs.

Gas Temperatures Versus Metal Temperature of Blades

In order to establish the metal temperature of the blades of the supercharger in terms of the temperature of the gas in the nozzle box, it was necessary to employ an indirect method. The procedure of attaching a thermocouple to one of the blades of the rotor and carrying the leads through a hollow shaft to a set of slip rings was considered. The closely spaced, thin-blade sections are not well-adapted for mounting a thermocouple, and it

was feared that the blade section might be damaged. However, this method has been employed successfully with larger rotors.

In the following procedure, powders and silver-brazing alloys of known melting characteristics were employed for indicating metal temperatures of the blades. Four blades, located 90 deg apart, were removed from a standard rotor of a Type B supercharger. The concave side of these blades was filled in with 25:20 chromium-nickel weld metal, in order to thicken up the section and provide body for an axial hole 0.136 in. diam. The blades were finished to approximately equal weights and reinstalled in the rotor. Capsules of 25:20 chromium-nickel alloy of the type shown in Fig. 6 were prepared for each of the modified blades. One end of the tubular capsule is threaded for screwing into the axial hole in the end of the blade. The space within the capsule is for holding either temperature-indicating powders or alloys of known melting characteristics. Centrifugal force tends to force the powder or alloy toward the closed end of the capsule. As long as the temperature-indicating material remains in the solid state, there is no tendency for it to pass through the small radial hole located near the closed end of the capsule. However, if the material is in the liquid or viscous state, it will be forced through the radial hole. Special precautions are necessary to prevent seizure between the threads of the capsule and blade section. This difficulty was obviated by preoxidizing the threads of the capsule before assembly.

Prior to conducting tests in the supercharger rotor, calibration tests were made by placing temperature-indicating powders and alloys in similar capsules mounted in steel blocks. These assemblies were heated in a muffle furnace at temperatures ranging from 1300 to 1450 F, and the behavior of the materials observed.

Tests were made at 1336 deg, 1386 deg, and 1420 deg F temperatures in the supercharger. In testing at each of these temperatures, a temperature-indicating powder of a different melting point was placed in the capsule of each of the four special blades. The testing temperatures are the average gas temperatures as described previously. Test results obtained with temperature-indicating powders are given in Table 6.

Temperature calibration tests also were made of three silver-brazing alloys, the alloys being placed in tubular capsules contained in steel blocks, as previously described for the temperature-indicating powders. Flow points and melting ranges for the three alloys are given in Table 7. Three test runs of the supercharger were made with each of these alloys contained in a blade. The fourth blade contained powder prepared from a 1300 F temperature pellet. The operating conditions of the supercharger were adjusted to give a temperature of 1320 F for the test run. Results obtained for the three alloys and the powder after 1 hr operation of the supercharger are given in Table 8.

Discussion

One would naturally inquire as to how the deformation of the several alloys tested as blades in the turbosupercharger compare

TABLE 6 RESULTS OF TESTS WITH TEMPERATURE-INDICATING POWDERS

	Temperature of tests— Gas temperature of supercharger: 1336 F		1386 F		1420 F	
Blade position in supercharger rotor	Designated pellet temperature, deg F	Condition of pellet powder after run	Designated pellet temperature, deg F	Condition of pellet powder after run	Designated pellet temperature, deg F	Condition of pellet powder after run
1	1250	(d)	1300	(d)	1300	(d)
2	1300	(d)	1350	(c)	1350	(d)
3	1350	(b) 1/32 in.	1400	(a)	1400	(c)
4	1400	(a)	1450	(a)	1450	(b) 1/32 in

Notes:
(a) Powder apparently unaffected.
(b) Powder shrank amount indicated but otherwise solid.
(c) Top and side crust remaining.
(d) All powder had disappeared from fixture

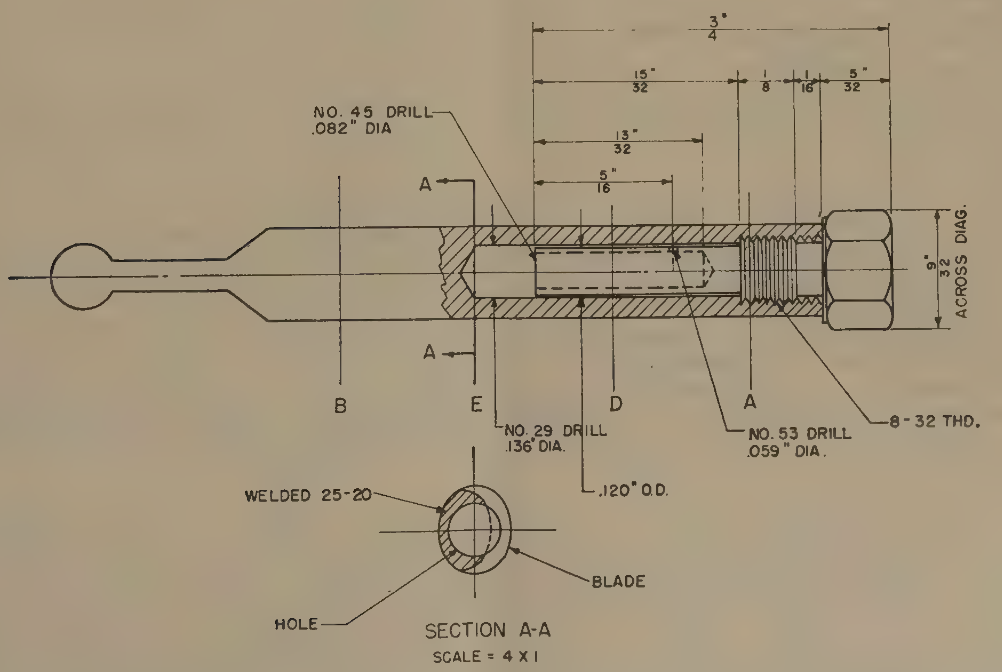

Fig. 6 Capsule Mounted in End of Turbine Blade for Holding Temperature-Indicating Powder or Alloy

TABLE 7 RESULTS OF TESTS WITH SILVER-BRAZING ALLOYS

Alloy designation	Flow point, deg F	Melting range, deg F
1	1272	1191–1272
2	1320	1280–1323
3	1345	1339–1345

TABLE 8 RESULTS OF TESTS WITH SILVER-BRAZING ALLOYS AND TEMPERATURE-INDICATING POWDER

Test run (1320 F)"	Temperature-indicating materials: Silver-brazing alloys 1	2	3	Pellet 1300F rating
1	Alloy melted and ran from radial hole	Alloy shrank but did not run out	Intact	Shrank slightly
2	Alloy melted and ran from radial hole	Alloy melted and ran from radial hole	Alloy shrank slightly	Shrank $^1/_{16}$ in.
3	Alloy melted down but did not run out	Alloy partly melted	Intact	Marked shrinkage, only crust remaining

with results obtained for laboratory test specimens. For purposes of this comparison test data for laboratory test specimens contained in the NDRC reports, particularly the final report dated January 21, 1946, were examined. The twelve test alloys were arranged in order of their rupture strength at 100 and 1000 hours at 1500 F temperature. Results were selected for those alloys which appeared to be most comparable with the blade materials as regards chemical composition and heat-treatment. For 100 hours fracture time the four alloys showing the highest rupture strengths were 61, 422-19 K-42-B, and 6059 in the order listed. At 1000 hours rupture time the corresponding alloys were: 61, 422-19, 6059, and vitallium. For both rupture periods 16:25:6 and 19:9 DL showed the lowest rupture strength of the series.

A similar comparison of creep-test data contained in the NDRC reports was made. The alloys were listed in order of their deformation rates for 1000 psi stress at 1500 F temperature at 1000 hours. The four alloys leading the list were: Refractaloy, 6059, 422-19, and 61. It is pointed out that three alloys, 61, 422-19, and 6059 appear in the top group by either method of comparison.

Results obtained for alloys listed in the supercharger showed the first four alloys to be, K-42-B, 61, N-155, and 422-19. Accordingly, K-42-B and N-155, showed to better advantage comparatively in the supercharger test than when tested as laboratory test specimens. Refractaloy and alloy 6059 which showed the maximum resistance to deformation at 1500 F in laboratory creep tests stood approximately in the middle of the group tested in the supercharger. The 16:25:6 and 19:9 DL alloys showed the minimum creep resistance in the supercharger test which is in line with both stress-rupture and creep test data for laboratory test specimens.

Summary

The need for test information concerning the high-temperature strength characteristics and stability of gas-turbine alloys beyond that provided by stress-rupture, creep, and gas-erosion tests is discussed. The practicability of testing a number of alloys in the form of blades in an aircraft turbosupercharger operated as a gas turbine is pointed out. By this procedure, blades of different alloys are subjected simultaneously to the combined effect of stress and erosion by hot combustion gases after the manner of a gas turbine. This combination of test conditions is not a[t]tained readily with the usual laboratory high-temperature testi[ng] facilities. An indirect method for evaluating blade-metal te[m]peratures, corresponding to gas temperatures in the nozzle cha[m]ber, is described.

Future experimentation is to include quick-starting tests of [the] supercharger, and the testing of a second rotor. The latter co[n]tains blades of fourteen alloys, including seven cast and seve[n] wrought materials. Most of these represent alloys of later d[e]velopment than those of the first rotor.

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The Applicability of Ceramics and Ceramals as Turbine-Blade Materials for the Newer Aircraft Power Plants

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Ceramic and ceramal materials have been investigated for use as turbine-blade materials for aircraft gas turbines. Tensile, flexure, thermal-shock, and oxidation data for these materials at temperatures up to 2400 F are presented. These data are discussed with respect to applicability for turbine use. Results of turbine-blade operation at the NACA Cleveland laboratory are given.

Introduction

The turbine blade is one of the most critical parts of a gas turbine. Although both the combustion chambers and the stator vanes operate in gas at higher temperatures than that passing the turbine blades, the combustor and stator components do not present great design problems because they are subjected principally to thermal stresses alone and because they can be cooled by circulation of a gas or a liquid. The turbine blade, on the other hand, is subjected to centrifugal, vibratory, and thermal stresses, and is more difficult to design with cooling provisions because of high speeds of rotation.

It is well known that an increase in temperature of a gas turbine results in an increase of thermal efficiency of the unit. Present-day inlet gas temperatures are about a maximum of 1650 F with occasional rises to as much as 2000 F (for example, during starting). This 1650 F gas temperature produces maximum turbine-blade temperatures of about 1500 to 1550 F. Increases in average gas temperature to 1950 F result in large increases in turbine thermal efficiency. The corresponding maximum metal temperature in the turbine blade would then be about 1800 F.

Tensile Properties

Table 1 lists strengths, taken from literature, of some typical commercial high-temperature alloys at 1500 and 1800 F. The short-time tensile strengths were determined in a standard tensile-test machine which mounts a furnace surrounding the test specimen. The furnace winding is such that a uniform temperature is maintained along the length of the specimen. The stress-rupture strengths given in Table 1 are the dead loads which may be supported by a specimen for 100 hr without fracture. Material A is a commercial alloy in use in gas turbines with a service life no more than adequate at present gas operating temperatures corresponding to a metal temperature of 1500 F. The stress-rupture strengths are more indicative of blade performance than the short-time tensile strengths.

Because a well-designed turbine blade is subjected mainly to centrifugal and thermal stresses, and only secondarily to vibratory stress, a useful criterion of turbine-blade materials is strength-weight ratio. Table 1 lists the ratios of the short-time tensile strengths and of the 100-hr stress-rupture strengths to the specific gravities of the corresponding metal relative to the

TABLE 1 ELEVATED-TEMPERATURE STRENGTHS OF COMMERCIAL HIGH-TEMPERATURE ALLOYS

Alloys	Specific gravity	1500° F: Short-time tensile strength (lb/sq in.)	1500° F: Tensile strength / R. S. G.[a]	1500° F: 100-hour stress-rupture strength (lb/sq in.)	1500° F: 100-hour stress-rupture strength / R. S. G.[a]	1800° F: Short-time tensile strength (lb/sq in.)	1800° F: Tensile strength / R. S. G.[a]	1800° F: 100-hour stress-rupture strength (lb/sq in.)	1800° F: 100-hour stress rupture strength / R. S. G.[a]
A	8.30	61,400	61,400	22,000	22,000	33,300	33,300	8,800	8,800
B	9.24	77,500	69,600	17,000	15,300	31,500	28,300	----	----
C	8.59	78,300	75,700	27,000	26,100	22,200	21,500	5,300	5,100
D	8.31	----	----	20,000	20,000	18,500	18,500	5,500	5,500
E	8.61	61,400	59,200	28,000	27,000	29,400	28,300	11,300	10,900
F	8.54	59,500	57,800	26,700	25,900	33,100	32,200	8,700	8,500
G	8.38	51,400	50,900	24,000	23,800	33,700	33,400	9,200	9,100
H	8.31	65,000	64,900	30,400	30,400	37,800	37,800	10,100	10,100

[a] Relative specific gravity based upon specific gravity of alloy A equal to 1.00.

[1] Aeronautical Research Scientist, Lewis Flight Propulsion Laboratory, National Advisory Committee for Aeronautics, Cleveland Airport. Mem. ASME.

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specific gravity of alloy A taken as unity. These values will be compared to similar values for ceramics and ceramals.

The first extensive attempt to evaluate ceramics as substitute materials in turbine-engine applications was made by Geller and co-workers at the National Bureau of Standards under sponsorship of the NACA. Creep and modified stress-rupture measurements were made on bodies formulated at the National Bureau of Standards after commercial ceramic bodies had been found lacking in desired properties (1).[2] Table 2 lists the 100-hr stress-rupture strengths of the best materials developed. The strength-weight ratios are given with the specific gravity again relative to alloy A as unity. The ceramics were fabricated either by slip casting or by hydrostatic pressing and subsequent firing. The alloys of best strength-weight ratio listed in Table 1 are given for purposes of comparison. Sillimanite, a commercial ceramic material, is also placed in the table because it was utilized as a turbine-blade material while the National Bureau of Standards program was in progress (2).

Another method of fabricating ceramic materials is that of hot-pressing. A number of bodies fabricated by this technique were obtained from the Norton Company of Canada. The bodies were boron carbide, 85 per cent boron carbide—15 per cent silicon carbide, magnesia, beryllia, zirconia (stabilized), zircon sand, titanium carbide, and zirconium carbide. It will be noted that carbides are termed ceramics, as well as the usual oxides and silicates. Borides, nitrides, and silicides, which are not mentioned in this paper, are also termed ceramics. Short-time tensile tests were made on these bodies, and the results are given in Table 3 (3). Strength-weight ratios are given relative to unit specific gravity of alloy A.

The maximum strength-weight ratio for the best commercial alloys is given for purposes of comparison. Because only stress-rupture values were available for National Bureau of Standards ceramics, strength-weight ratios are given for these materials using stress-rupture strengths instead of tensile strengths. The strength properties of the hot-pressed materials are lower, in general, than the best National Bureau of Standards materials at 1800 F, but are probably better at 2200 F on the basis of indirect tensile data on the National Bureau of Standards bodies.

A number of ceramals (combinations of ceramics and metals that are pressed and sintered or hot-pressed) were evaluated in elevated-temperature tensile tests. These ceramals, obtained from the Kennametal Corporation, contained titanium carbide as the ceramic, and cobalt as the metal in varying proportions. Specimens were 0.505 in. diam at the test section and were of the shape recommended in reference (4). The results of these evaluations are given in Table 4 (5). The values in Table 4 are either averages for two specimens, or results of single tests. It will be noted that the strength-weight ratio is a maximum for the 20 per cent cobalt composition. The probable reason for the peak is that an optimum amount of metal serves as binder in combination with the ceramic, and that additional metal provides weaker material (the metal), in the interstices between the bonded carbide particles. This optimum percentage of metal should vary inversely as the size of ceramic particles because of the change in volume-area ratio of the particles with size.

A comparison of all materials discussed may be made from Table 4. It can be seen that at a temperature of 1800 F, the best ceramal is competitive in tensile strength with the best commercial heat-resistant alloy and with the best National Bureau of Standards ceramic, although inferior to the best hot-pressed ceramic; at 2200 F, the best ceramal is superior to the best commercial heat-resistant alloy and is competitive with the best hot-pressed ceramic.

[2] Numbers in parentheses refer to the Bibliography at the end of the paper.

TABLE 2 ELEVATED-TEMPERATURE STRENGTHS OF CERAMIC BODIES DEVELOPED AT THE NATIONAL BUREAU OF STANDARDS[a]

Material Designation	Abbreviated Composition	Specific Gravity	1500° F		1800° F		1900° F	
			100-hour stress-rupture strength[b] (lb/sq in.)	100-hour stress-rupture strength / R.S.G.[c] (lb/sq in.)	100-hour stress-rupture strength (lb/sq in.)	100-hour stress-rupture strength / R.S.G.[c] (lb/sq in.)	100-hour stress-rupture strength (lb/sq in.)	100-hour stress-rupture strength / R.S.G.[c] (lb/sq in.)
4811	High Beryllia	3.0	14,000	38,700	17,000	47,000	16,000	44,200
151	Beryllia and Zirconia	3.8	13,000	28,400	17,000	37,100	12,000	26,200
353	High Zirconia	4.4	12,000	22,600	18,000	34,000	10,000	18,900
358	High Zirconia	4.9	12,000	20,300	17,000	28,800	8,000	13,500
16021	High Beryllia	3.0	12,000	33,200	6,000	16,600	----	----
163	High Zirconia	4.4	11,000	20,700	16,000	30,200	10,000	18,900
Sillimanite[d]	Aluminum silicate	2.8	11,500	34,000	5,500	16,300	3,500	10,400
Alloy E[e]	----	8.61	28,000	27,000	11,300	10,900	----	----
Alloy H[e]	----	8.31	30,400	30,400	10,100	10,100	----	----

[a] Data obtained from modified stress-rupture tests.
[b] Values from early research and are probably too low.
[c] Relative specific gravity based upon specific gravity of alloy A (8.30) equal to 1.00.
[d] Not a development of the National Bureau of Standards.
[e] Not a ceramic. Values from Table 1.

TABLE 3 SHORT-TIME TENSILE STRENGTHS OF HOT-PRESSED CERAMIC BODIES

Composition	Apparent density (grams/ml)	1800° F		2200° F	
		Tensile strength (lb/sq in.)	Tensile strength / R.S.G.[a] (lb/sq in.)	Tensile strength (lb/sq in.)	Tensile strength / R.S.G.[a] (lb/sq in.)
B_4C	2.50	22,600	75,000	----	----
TiC	4.74	17,200	30,100	9,400	16,400
ZrC	6.30	14,450[b]	19,000[b]	15,800	20,800
SiC + 15% B_4C	3.00	9,950	27,500	7,100	19,600
Zircon Sand	4.54	8,700	16,250	3,600	6,600
BeO	3.0	6,200	17,200	----	----
Stabilized ZrO_2	5.80	6,750	9,650	----	----
MgO	3.39	3,100	7,600	----	----
Alloy H[c]	8.31	37,800	37,800	----	----
NBS Ceramic No. 4811[d]	3.0	19,000	52,600	4,000[e]	11,000

[a] Relative specific gravity based on specific gravity of alloy A equal to 1.00.
[b] Probably low values.
[c] Not a hot-pressed body. Values obtained from Table 1.
[d] Not a hot-pressed body.
[e] Data from a 4000-psi stress-rupture test at increasing temperature—a minimum value.

TABLE 4 SHORT-TIME TENSILE STRENGTHS OF TITANIUM CARBIDE-COBALT CERAMALS

Composition	Density (grams/ml)	1800° F		2200° F	
		Tensile strength (lb/sq in.)	Tensile strength / R.S.G.[a] (lb/sq in.)	Tensile strength (lb/sq in.)	Tensile strength / R.S.G.[a] (lb/sq in.)
TiC[b]	4.74	17,200	30,100	9,400	16,400
TiC + 5% Co	5.06	22,600	37,100	9,800	16,100
TiC + 10% Co	5.07	24,600	40,400	14,400	23,600
TiC + 20% Co	5.37	34,600	53,500	13,200	20,400
TiC + 30% Co	5.61	22,600	33,400	14,700	21,700
Alloy H[d]	8.31	37,800	37,800	----	----
N.B.S. Ceramic No. 4811[d]	3.0	19,000	52,600	4,000[e]	11,000
B_4C[b]	2.52	22,600	74,200	----	----
ZrC[b]	6.31	14,450[c]	19,000[c]	15,850	20,850

[a] Relative specific gravity based upon specific gravity of alloy A equal to 1.00.
[b] This body was hot-pressed. Data from Table 3. Not a ceramal.
[c] Probably a low value.
[d] Not a ceramal.
[e] Data from a 4000-psi stress-rupture test at increasing temperature—a minimum value.

Bending Properties

The tranverse flexural strengths of a number of ceramals were determined at elevated temperature instead of the tensile properties, because of ease of evaluation of flexure properties. The specimens were machined in the form of rectangular bars $^1/_2$ in. in width, $^1/_4$ in. deep, and from 2 to 4 in. in length. A three-point loading system was employed. Temperatures of 1600, 2000, and 2400 F were fixed as evaluation temperatures. The specimens evaluated were composed of titanium carbide and varying proportions of cobalt, of tungsten, or of molybdenum. The objects of these tests were to determine if an optimum composition exists for each pair of component materials, and to determine the effect of refractoriness of metal binders on the strength of ceramals at high temperatures. If it is assumed that there is an optimum quantity of metal binder that is just adequate to coat all ceramic particles, a deficiency of metal would result in an incomplete bond, and an excess of metal provides a superfluity of a material weaker in physical properties than the remainder of the body.

The results of these flexure evaluations performed in inert atmosphere (5) are given in Fig. 1, in which percentage metal by weight is plotted against modulus of rupture (calculated maximum stress in outer fibers of specimen at fracture) with temperature of test as a parameter. At the lower temperatures, 20 per cent cobalt appears to be the optimum proportion, but the optimum quantities of molybdenum and of tungsten are not consistent at the different temperatures, although the tungsten ceramal appears to peak at 10 per cent metal content. At the highest temperature of evaluation, 2400 F, the molybdenum body is strongest, the tungsten less strong, and the cobalt weakest. At the lower temperatures, 1600 F and 2000 F, the cobalt-bonded titanium carbide is strongest.

An approximate correlation can be made between modulus of rupture and short-time tensile strength based on results for brittle materials at a number of laboratories. The short-time tensile strength is usually found to be about 0.4 to 0.6 of the modulus of rupture. Assuming that the short-time tensile strengths are 0.5 of the modulus of rupture, the values of strength and strength-weight ratio (with the specific gravity of alloy A as unity) are given in Table 5 for these materials. These values were

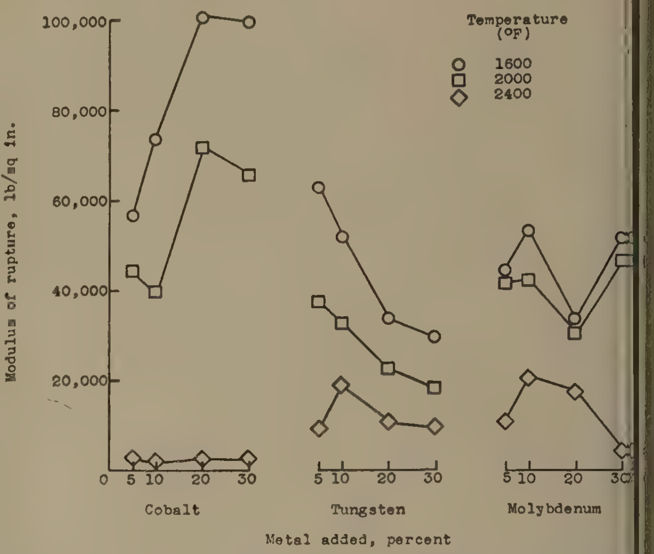

Fig. 1 Effect of Composition on Modulus of Rupture Strengths of Some Titanium-Carbide-Base Ceramals

obtained by interpolation from Fig. 1. For comparison purposes, similar values are given for alloy A and for the best ceramics.

TABLE 5 TENSILE STRENGTHS OF TITANIUM-CARBIDE-BASE CERAMALS

Alloy	Density (grams/ml)	1800° F[a] Tensile strength[b] (lb/sq in.)	1800° F[a] Tensile strength / R.S.G.[c] (lb/sq in.)	2200° F[a] Tensile strength[b] (lb/sq in.)	2200° F[a] Tensile strength / R.S.G.[c] (lb/sq in.)
TiC + 5% Co	5.06	25,900	42,400	13,500	22,100
TiC + 10% Co	5.07	28,200	46,200	11,100	18,200
TiC + 20% Co	5.37	43,600	67,100	20,800	31,900
TiC + 30% Co	5.61	42,200	62,000	18,100	26,700
TiC + 5% Mo	5.12	21,700	34,900	14,800	23,800
TiC + 10% Mo	5.06	24,600	40,300	16,200	26,600
TiC + 20% Mo	5.24	16,400	26,000	12,200	19,400
TiC + 30% Mo	5.77	24,900	35,500	15,100	21,600
TiC + 5% W	5.14	25,300	40,700	12,100	19,500
TiC + 10% W	5.22	20,600	32,700	12,900	20,400
TiC + 20% W	5.31	14,300	22,300	8,500	13,300
TiC + 30% W	5.81	11,900	17,000	7,000	10,000
Alloy H[d]	8.31	37,800	37,800	----	----
NBS 4811[e]	3.0	19,000	52,600	4,000[f]	11,000[f]
B_4C[e]	2.52	22,600	74,200	----	----
ZrC[e]	6.31	14,450	19,000	15,850	20,850

[a] Tensile-strength values interpolated from data at 1600, 2000, and 2400 F.
[b] Tensile strength = modulus of rupture/2.
[c] Relative specific gravity based upon specific gravity of alloy A equal to 1.00.
[d] Data obtained from Table 1.
[e] Data obtained from Table 3.
[f] Data from a 4000-psi stress-rupture test at increasing temperature—minimum value.

Creep

Creep data are important because they permit estimation of the time that turbine blades of a given material can be operated without excessive deformation. Such data are available, however, only for alloys and for the National Bureau of Standards ceramics at the present time. These data are given in Table 6 in terms of stress for a given creep rate and ratio of stress to specific gravity with the specific gravity of alloy A as unity. It is known that alloy B is only slightly better than adequate for operation at a metal temperature of 1500 F. The creep rates of the best ceramics at temperatures of 1800 and 1900 F are seen to be adequate, in general, when compared to the creep rate of alloy B at 1500 F.

Thermal Shock

Thermal shock of turbine blades occurs when a turbine is started or stopped. The temperature gradients produced may be high enough to crack metals as well as ceramics and ceramals, but usually a number of shock cycles are required to produce fracture. The failure typical of ceramic and ceramal specimens of turbine-blade dimensions is a fracture through the body of the material rather than a spalling of the surface.

The resistance to failure by thermal shock is poorer, in general, for oxide-base materials than for carbide-base materials, chiefly because of the higher thermal conductivity of the carbides. Most metals are very resistant to thermal shock because they possess good thermal conductivity and high ductility, which permit large values of allowable strain before fracture.

Norton (6) has summarized the fundamental factors involved in failure by thermal stress. Briefly, the properties of materials at elevated temperatures required for maximum resistance to thermal shock are as follows:

1 High tensile strength.
2 High thermal conductivity.
3 Low specific heat on a volume basis.
4 Low effective modulus of elasticity (ratio of stress to strain at fracture).
5 Low coefficient of thermal expansion.
6 Low emissivity.
7 Low exterior conductivity (heat loss by convection at the surface).

Data were available for a number of materials on strength, conductivity, effective modulus, and expansion, so that a correlation of these factors with thermal shock resistance (7) could be attempted.

The resistance to failure from thermal shock was determined by heating circular disks 2 in. diam and $^1/_4$ in. thick to a given temperature and quenching them in a moving stream of cooling air. If a specimen withstood 25 cycles of shock at a given temperature, the test was continued at the next higher temperature for another 25 cycles, to a maximum temperature of 2400 F.

The parameter used in evaluation was

$$\frac{(K)(TS)}{(\alpha)(E)}$$

where

K = thermal conductivity
TS = tensile strength
α = coefficient of thermal expansion
E = effective modulus of elasticity

A large value of this parameter is indicative of good resistance to thermal shock. Table 7 lists the correlation obtained for a titanium carbide-cobalt ceramal, for magnesia, for titanium carbide, for beryllia, for zircon sand, and for stabilized zirconia.

TABLE 6 CREEP STRENGTHS FOR COMMERCIAL HIGH-TEMPERATURE ALLOYS AND CERAMICS

		1500° F		1600° F		1700° F		1800° F	
Alloy	Specific gravity	Stress for minimum creep rate of 0.0001 percent per hour (lb/sq/ in.)	Stress for minimum creep rate of 0.0001 percent per hour / R. S. G.[a] (lb/sq in.)	Stress for minimum creep rate of 0.0001 percent per hour (lb/sq in.)	Stress for minimum creep rate of 0.0001 percent per hour / R. S. G.[a] (lb/sq in.)	Stress for minimum creep rate of 0.0001 percent per hour (lb/sq in.)	Stress for minimum creep rate of 0.0001 percent per hour / R. S. G.[a] (lb/sq in.)	Stress for minimum creep rate of 0.0001 percent per hour (lb/sq in.)	Stress for minimum creep rate of 0.0001 percent per hour / R. S. G.[a] (lb/sq in.)
A	8.30	7,600	7,600	----	----	----	----	----	----
B	9.24	2,800	2,500	----	----	----	----	----	----
C	8.59	11,500	11,000	5,800[b]	5,600[b]	----	----	----	----
D	8.31	9,600	9,500	----	----	----	----	----	----
E	8.61	13,700	13,200	11,700	11,300	----	----	----	----
F	8.54	13,000	12,600	----	----	----	----	----	----
G	8.38	11,700	11,600	9,000	8,900	----	----	----	----
H	8.31	13,200	13,200	11,600	11,600	----	----	----	----
NBS 4811	3.0	----	----	14,000	38,700	14,000	38,700	10,000	27,600
NBS 353	4.4	----	----	----[c]	----[c]	17,000	32,100	5,000	9,400
NBS 358	4.9	----	----	----[c]	----[c]	----	----	5,000	8,500
NBS 16021	3.0	----	----	13,000	35,900	12,000	33,200	5,000	13,800
NBS 163	4.4	----	----	----[c]	----[c]	16,000	30,200	4,000	7,500

[a] Relative specific gravity based upon specific gravity of alloy A equal to 1.00.
[b] This alloy is used in the as-forged condition which may account for the rapid decrease in the stress from the 1500 F to 1600 F test.
[c] Creep rate negligible for stresses at which tested.

Alloy A is presented for comparison, although no thermal shock data were obtained for it. The correlation is seen to be good and shows the marked influence of thermal conductivity for the carbides.

A number of other materials were evaluated in thermal shock without obtaining data for correlation purposes. These materials are given in Table 8 in order of descending resistance to thermal shock. In general, the carbide-base materials are superior to the oxide-base materials, provided the tensile strengths at elevated temperature are adequate.

The effect of time at stress and temperature on the alteration of microstructure and concomitant physical properties has been neglected in this investigation because of the large expenditure of time and specimens which would have been required to obtain such long-time properties as 100-hr stress-rupture strengths. This factor should be taken into account when conducting a complete investigation on a material thought to be suitable for turbine-blade use, in order to evaluate the deterioration of tensile properties with time.

Another factor which has been neglected is the cumulative effect of successive thermal shocks on the tensile properties of the materials. It is probable that failure in a turbine blade might occur from the action of centrifugal stress after the material had been weakened by shocks. The data required for analysis of possible failure by this mechanism would be tensile strength after a given number of thermal shocks.

TABLE 7 CORRELATION OF MATERIAL PROPERTIES WITH RESISTANCE TO FRACTURE BY THERMAL SHOCK

Order of merit of materials evaluated in thermal shock	Thermal shock cycles before failure, Temperature (°F) 1800	2000	2200	2400	Coefficient of thermal expansion for temp. range 80 - 1100°F (in./in./°F) $c \times 10^{+6}$	Thermal conductivity (Btu/(ft²)(hr)(°F/in.) K	Effective modulus of elasticity at 1800° F (lb/sq in.) $E \times 10^{-7}$	Tensile strength at 1800° F (lb/sq in.) S	$\frac{KS}{cE}$
Alloy A	[a]				8.24	140	.10	33,300	566,000
80% TiC + 20% Co	25	25	25	25[b]	5.5	240	6.0	34,600	25,200
TiC	25	25	25	17	4.56	240[c]	6.0[c]	17,200	15,100
BeO	25	3			5.1	104	4.28	6,200	2,950
$ZrSiO_4$	1				2.51	11.6	2.4	8,700	1,700
MgO	1/2				7.69[d]	16-40	1.24	3,100	520-1300
94% ZrO_2 + 6% CaO	0				5.53	14.3[e]	2.5[e]	6,750	700

[a] Not yet evaluated, but probably best of all materials given.
[b] No failure.
[c] Value for 80 per cent TiC + 20 per cent Co.
[d] Temperature range 80 to 925 F.
[e] Value for ZrO_2.

Corrosion

The principal form of corrosion to be considered in gas turbines operating on fuel of low sulphur and negligible tetraethyl lead content is oxidation. Oxidation has no effect on oxides, except for the possibility of forming oxides of higher oxygen content. It is necessary, however, to determine the oxidation rates of carbide-containing materials as well as ceramals, in which the metal may be oxidized.

Oxidation rates have been determined for some carbides (8) and carbide-base ceramals. These rates are given in Table 9. In analyzing the rates, the quality of the oxidation product must be considered. For example, the oxides formed on a turbine blade of 20 per cent cobalt-titanium carbide ceramal were capable of withstanding operation at high gas temperatures and rotative speeds when as much as $^1/_{16}$ in. thick.

The oxidation rates shown for the carbide-type ceramals at moderate operating temperatures are not excessive, but those at gas temperatures of 2100 to 2200 F are so high that life of a turbine blade probably would be limited by oxidation to about 10 hr. The resistance to oxidation possibly may be improved by additions of other basic materials so that a different oxide, more resistant to protracted oxidation, can be formed.

A necessary step in the search for oxide products which are resistant to further oxidation is the identification of the oxide produced. This identification is readily performed by a combination of x-ray diffraction and wet chemical analysis. X-ray diffraction

TABLE 8 THERMAL-SHOCK EVALUATION OF CERAMICS AND CERAMALS

Composition	Specimen No.	Number of cycles to failure 1800° F	2000° F	2200° F	2400° F
80/20% TiC/Co	3D13	25	25	25	25[a]
	3D14	25	25	25	25[a]
100% TiC	3A7	25	25	25	14
	3A12	25	25	25	21
85/15% SiC/B_4C	6C7	25	25	2	--
	6C12	25	6	--	--
100% BeO	1A1	25	3	--	--
100% ZrC	4A5	22[b]	--	--	--
100% $ZrSiO_4$	7A10	1	--	--	--
	7A11	1/2	--	--	--
100% B_4C	5A11	1/2[c]	--	--	--
	5A12	1/2[c]	--	--	--
100% MgO	2A7	1/2	--	--	--
ZrO_2 (Stabilized)	8A11	0	--	--	--
	8A12	0	--	--	--

[a] No failure.
[b] Excessive oxidation, sample fell out of holder.
[c] Values are believed to be low.

TABLE 9 OXIDATION RATE OF SOME CARBIDE-BASE MATERIALS

Composition	Oxidation Penetration in 50 hours (in.)			Oxidation Penetration in 100 hours (in.)		
	Temperature					
	1625° F	1800° F	2000° F	1625° F	1800° F	2000° F
26%Cr - 20%Ni[a] (1)	0.00004[b]	0.0001	0.0002	0.00008[b]	0.0002	0.0004
25%Cr - 12%Ni[a] (1)	.000025[b]	.0001	.0003	.00005[b]	.0002	.0006
85%SiC - 15%B_4C	.0007	---	.003[c]	.0014	---	.006[c]
TiC	.0024[c]	---	---	.004[c]	---	---
TiC + 5% W	.0015	.0037	---	.0020	.0043[c]	---
TiC + 10% W	.0011	.0040[c]	---	.002	.0048[c]	---
TiC + 20% W	.0044	.0047	---	.0078	---	---
TiC + 30% W	.0073	.0093[c]	---	---	---	---
TiC + 5% Co	.0023	.0062	.0165	.0035	.010	.024[c]
TiC + 10% Co	.0028	.0058	.028	.0042	.0095	.044
TiC + 20% Co	.0027	.008	.028	.0044	.011	.045[c]
TiC + 30% Co	.0032	.0092	.022[c]	.0068	.0156	---
TiC + 5% Mo	.0025	.014[c]	.080[c]	.0065	.0275[c]	---
TiC + 10% Mo	.0030	.0245[c]	---	.0075	.0475[c]	---
TiC + 20% Mo	.0025	.050[c]	---	.0050	---	---
TiC + 30% Mo	.0205[c]	---	---	---	---	---
B_4C	.003	---	.11[c]	.006[c]	---	.22[c]
ZrC	3.43[c]	---	---	6.86[c]	---	---

[a] Not a carbide-base material.
[b] 1600 F data.
[c] Extrapolated data.

alone is suitable, provided there is no possibility of the formation of isomorphs among which diffraction cannot distinguish. By this means it was found that the oxide formed on the 20 per cent cobalt titanium-carbide ceramal was composed of two layers, one of TiO_2 (rutile), and one of $CoTiO_3$. Correlation of oxidation rate, consequently, should be made with constitution of the oxide coating in order to determine which additions should be made to the basis material for best oxidation resistance.

Thermal Conductivity

Thermal conductivity, in addition to influencing resistance to thermal shock, has a marked effect on the operating temperature of turbine blades. The heat from combustion gases is communicated to the blade which in turn loses heat to the cooler turbine wheel. The high conductivity of some carbide-base materials exceeds that of most commercial heat-resistant alloys and is, in general, far higher than oxide-type ceramics. A simple experiment in which alloy, ceramic, and ceramal blades are mounted in a wheel and passed through the flame from a gas torch for a given time reveals that the ceramic is white hot, the alloy bright red, and the ceramal a dull red. The same order of temperature would prevail in a turbine in which, for a given gas temperature, the ceramic would run hottest and the carbide-type ceramal coolest. Thus a designer must employ different temperatures of evaluation in analyzing the possible use of different materials as turbine blades. Table 10 presents data from literature on thermal conductivities of several materials.

Turbine-Wheel Evaluation

A number of turbine wheels have been run either completely or partly bladed with ceramic or ceramal blades. One wheel (2) was operated with sillimanite blades at low rotative speeds at gas temperatures up to 1600 F. Failure occurred at a comparatively high speed, 10,000 rpm, at 500 F gas temperature because of stress concentration between wheel and blades. A redesigned blade was operated for 38 hr at temperatures as high as 1725 F and speeds up to 8700 rpm. The difficulties encountered with this unit were primarily problems of mechanical design so that

TABLE 10 COEFFICIENTS OF THERMAL CONDUCTIVITY OF CERAMICS, CERAMALS, AND METALS

Material	Coefficient of thermal conductivity Btu/(ft^2)(hr)(°F/in.)				
	392° F	1112° F	1600° F	2000° F	2400° F
Silicon carbide, recrystallized	---	---	115	95	81
Zirconia	---	---	13.0	13.7	14.3
Alumina, fused	---	---	26.7	28.4	30.0
Silica	---	---	12.4	14.0	15.7
80% Titanium carbide 20% Cobalt	240	---	---	---	---
Alloy A	---	142	---	---	---
Alloy B	---	114	---	---	---

the potentialities of the ceramic could not be determined definitely.

Blades of similar design were fabricated from National Bureau of Standards 4811 material. The results (9) indicate that the material is suitable for operation at least at speeds of 14,000 rpm and gas temperatures of 1800 F. A severe thermal shock set up by failure of the air supply caused fracture of the blades.

Blades of another design were fabricated by the Kennametal Corporation from 20 per cent cobalt-titanium carbide in such manner as to be interchangeable with metal blades in a turbine. Three ceramal blades and 139 blades of alloy A were successfully operated (10) at gas temperatures up to 2000 F and speeds up to 15,000 rpm. Almost simultaneously with increase of speed to 17,500 rpm all ceramal blades failed at the root where material temperatures are very low. The probable cause of the failures was resonant vibration at the particular operating speed, although a contributing cause probably was damage done to the blade roots when the wheel metal was peened against the blade roots for retaining purposes.

Another installation of three blades of 20 per cent cobalt-titanium carbide ceramal was made, this time without peening the wheel metal against the blades. In addition, the blade roots were plated with copper in order to reduce possible localized stresses produced by bearing against the wheel. Twelve new control blades of metal (alloy A) and the three ceramal blades were operated at varying conditions including $2^1/_4$ hr at a gas-inlet temperature of 2200 F and a wheel speed of 15,000 rpm. Up to this point 50 per cent of the metal control blades had failed. During an overhaul for replacement of a failed metal blade, one ceramal blade was inadvertently fractured by being struck. Operation was resumed at 2200 F gas temperature and 15,000 rpm. After $3^1/_2$ more hours of operation at this condition, during which 2 more control blades failed, a piece of the wheel rim holding one ceramal blade pulled apart from the wheel proper. The direct cause of the failure is believed to be the high temperature at this point on the wheel caused by the high thermal conductivity of the carbide-type ceramal. Excessive creep of the rim had been noticed in previous runs, but replacement of the wheel had not been made for fear of damaging the ceramal blades. The third ceramal blade failed simultaneously with the wheel failure in what may have been a normal failure. It is difficult to state that any of the ceramal blade failures was normal, because of the events surrounding their destruction. All that can be concluded is that one ceramal blade outlasted 50 per cent of the metal blades, and one ceramal blade outlasted 67 per cent of the metal blades prior to failures that were not caused by failure of the blade material. The third ceramal blade outlasted 67 per cent of the metal blades when it failed, perhaps normally, simultaneously with a minor wheel failure.

It is apparent that carbide-type ceramals present wheel-cooling problems because the heat transfer from the combustion gas to the wheel rim is large. This problem probably can be solved by the use of better wheel materials or by added wheel cooling.

Conclusion

A number of ceramics and ceramals have demonstrated their excellent tensile properties at elevated temperature. Carbide-base materials possess good thermal shock resistance and operate cooler than most high-temperature alloys or oxide-base materials, although they may present oxidation problems and difficulties with wheel cooling. Both ceramics and ceramals have operated as blades in gas turbines at temperatures above those in service use with alloy blades, although speeds were lower. Lives of ceramic and ceramal materials are still short, primarily because of mechanical design problems which cannot be anticipated prior to research evaluation. Additional research is required on such factors as elevated-temperature fatigue properties before ceram-ics and ceramals may be considered competitive with heat-re-sistant alloys except for short-time operation.

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Discussion

J. W. Freeman.[3] The author is to be complimented on a thorough and complete coverage of a new and interesting de-velopment in structural materials. He is indeed fortunate to have the opportunity to be in a position to work with these new materials which show so much promise for extreme temperature service.

The presentation was so thorough that it is difficult to discuss the paper without simply raising questions which the workers in the field have not yet had time to solve. There are three questions, answers to which would be of interest:

1 What was the type of wheel failures with the ceramal blades?

2 Ceramics and ceramals present problems of mechanical design for the utilization of brittle materials. Is the testing of ceramic and ceramal-type blades in an engine designed for metals satisfactory for these brittle materials?

3 The data infer that ceramics and ceramals have properties superior to metals at the more elevated temperatures. Is it therefore not surprising that the metal blades performed so well at the extreme temperatures in comparison to the TiC-20 per cent Co ceramal blades?

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Author's Closure

It is difficult to specify unequivocally the nature of the single-wheel failure that has taken place. The ceramal blade funneled more heat to its root than did the alloy blades to their roots, consequently overheating the failure zone. Deformation had previously been noted in this region also. It may be presumed that the failure was of a stress-rupture type because no signs of fatigue nor excessive corrosion were displayed by the fractured surface.

In answer to Dr. Freeman's second question, the author thinks that each ceramal should be considered as a separate case for design especially at the root fastening. In general, however, the ceramals do not possess as high strength at the roots as the alloys because the root operates cooler than the airfoil section and the ceramals are chiefly advantageous at the higher temperatures. On the other hand, the lower densities of most ceramals reduce the centrifugal stress proportionately. The net result is that the root size may have to be increased for the heavier ceramals or remain unchanged for the lighter. Special precautions should be taken not to injure the blade root in mounting it in the wheel. The root may have to be protected by a thin coat of a soft buffer metal from compressive fracture if the wheel is forced against the blade because of thermal expansion.

The results presented on ceramal turbine blades are first results. Of all ceramal blades evaluated, only one can be considered as a possible normal failure. For these reasons, it is preferable to await additional evaluations before commenting on whether the potentialities of the carbide ceramals have been assessed to their fullest.

Changes in Internal Damping of Gas-Turbine Materials Due to Continuous Vibration

By G. B. Wilkes, Jr.,[1] Lynn, Mass.

A pneumatically driven elevated-temperature fatigue machine and its control are described briefly. The use of this machine to determine qualitatively the initial damping as well as the changes in damping of the test specimen during vibration are discussed. The variation in high-stress initial damping versus temperature has been determined qualitatively on four high-temperature alloys. The elevated-temperature damping has been checked qualitatively during continuous vibration at high stresses. Two materials with extremely high initial damping tended to lose a large portion of it with time. The other materials showed damping changes of as high as 25 per cent with some increasing and some decreasing. Most of the data were taken below the fatigue limit and at temperatures of 1200 F, 1350 F, and 1500 F.

Introduction

During the course of setting up and operating a pneumatically driven fatigue machine at high temperatures, it was noticed that, with constant air pressure (driving force), the amplitude of vibration, and hence stress, did not necessarily remain constant. In one case, the pressure required to maintain constant stress dropped by a ratio of 5 to 1 after only a few hours' vibration. This naturally provoked considerable interest among those concerned with the fatigue problem in turbine buckets, and a series of tests were made to determine, qualitatively, the damping characteristics of several materials during the course of vibration. The main objective of these tests was to evaluate the importance of the change in damping as well as the accuracy of the method used to determine it. Therefore these tests do not give a complete story, but do give a strong indication of what appears to be an important property of some metals.

Test Apparatus

Since the work about to be described was carried out on a new piece of apparatus, a brief description of it will be given, although the designer has previously reported it.[2] It is a vibrating-cantilever fatigue machine, employing compressed air for driving purposes. The top or free end of the bar holds the armature which consists of two opposed pistons. These face two nozzles which are the two ends of a variable-length piece of tubing. The driving air is fed into the center of the tube. The length of the tube or air column is so adjusted that its length is either one half or one and one-half times the wave length corresponding to the natural frequency of the bar. Under these conditions, the air column and bar form a resonant system with the vibrating air column supplying a driving impulse to the bar every half cycle. Fig. 1 shows the machine with the various parts identified. The bar itself is held in a heavy chuck which extends part way up into the furnace. The temperature is controlled by a thermocouple fastened to the bar at the point of maximum stress. Even though the furnace is quite short, the stress gradient in the bar is sufficiently steep to give essentially constant temperature over the highly stressed section. The small coil and magnet shown on one nozzle is the source of energy used for amplitude-control purposes.

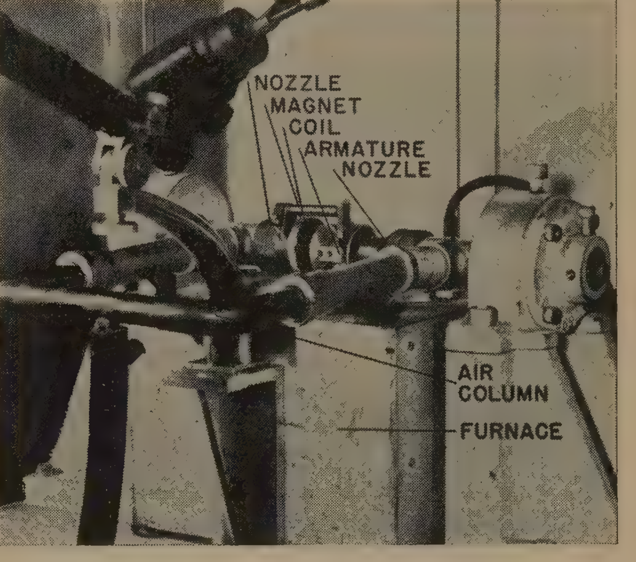

Fig. 1 Pneumatic Fatigue Machine

Fig. 2 shows the amplitude-control apparatus. The output of the coil and magnet is rectified and applied to a mirror galvanometer with suitable shunting. A light beam is reflected from the mirror to a photoelectric cell, which in turn operates a relay and a motor-driven air valve on the machine. Suitable limit switches shut down the entire machine in the event of bar failure or failure of any part of the control system. The filter is necessary to block any 60-cycle pickup from the furnace which would affect the control.

Damping data are obtained by measuring the steady pressure in the air column near the nozzle. This pressure is admittedly supplying energy to a number of places besides the internal damping of the test bar, but the author believes that the results which follow will support the contention that material damping changes are reflected in pressure changes. Owing to losses, other than damping of the test specimen, whose magnitudes are not known, any damping-capacity data taken in this manner should be considered as more qualitative than quantitative. The pressure in the air column is composed of a pulsating as well as a steady component. In order to determine the reliability of the steady component alone for power measurements, one bar

[1] Metallurgical Engineer, General Electric Company.

[2] "Pneumatic Fatigue Machines," by F. B. Quinlan, Proceedings of the American Society for Testing Materials, vol. 46, 1946, pp. 846–850.

Contributed by the Joint Committee on Effect of Temperature on the Properties of Metals and the Gas Turbine Power and Metals Engineering Divisions and presented at the Annual Meeting, New York, N. Y., November 28–December 3, 1948, of The American Society of Mechanical Engineers.

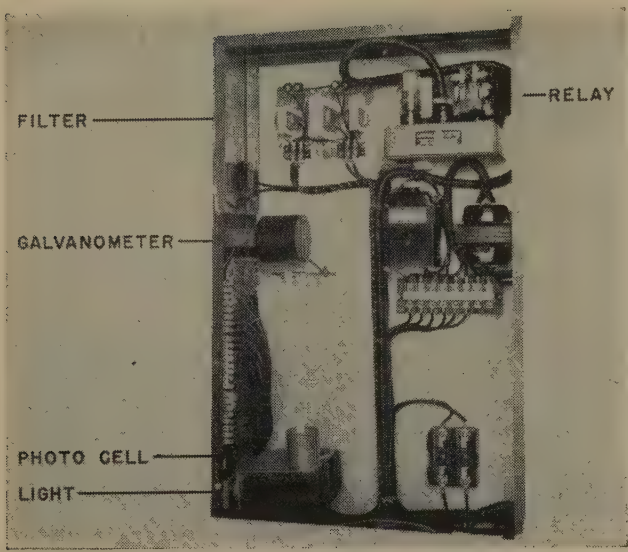

Fig. 2 Automatic-Control Panel

was run while measuring both components at the same time. The relation between them was found to be very close to linear at all pressures. The phase angle between the pulsating pressure component and the output of the control coil (motion of the bar) indicated that the system was tuned to give approximately maximum power.

Experimental Procedure

In order to obtain a general idea of the damping properties of alloys for high-temperature service, four alloys were chosen whose compositions and heat-treatments are shown in Table 1. Two more or less iron-base alloys (Timken 16–25–6 and N-155), one cobalt-base alloy (S-816), and one nickel-base alloy (Inconel X) were picked as representative of the field. Three test bars of each material were made, as shown in Fig. 3. There are several disadvantages of a rectangular bar, but since it was found quite difficult to get a round bar to vibrate in the proper plane, a rectangular bar became a necessity. The natural frequency of these bars ranged from about 150 cycles per sec (cps) at room

TABLE 1 COMPOSITION AND HEAT-TREATMENT OF ALLOYS USED IN TESTS

Composition

	Fe	Cr	Ni	Mo	W	Co	C	Cb	Ti
S-816	2	20	20	4	4	45	0.4	4	..
Inconel X	6	15	75	..	..	..	0.1	1	2
N-155	33	21	19	3	2	20	0.2	1	..
Timken	52	16	25	6	..	..	0.1	..	..

Treatment

S-816	2150 F—1 hr	1400 F—16 hr	
Inconel X	2100 F—3 hr	1550 F—24 hr	1300 F—16 hr
N-155	2175 F—1 hr	1400 F— 4 hr	
Timken	Cold-drawn bar stock		

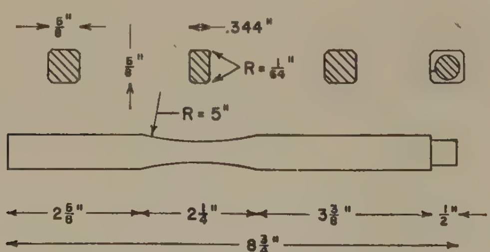

Fig. 3 Rectangular Fatigue Bar

temperature to 130 cps at 1500 F. For the initial damping data, i.e., before any prolonged vibration, one bar of each material was run in each of three machines in order to best rule out any pressure changes due solely to the characteristics of a particular machine. A pressure-versus-stress curve was obtained on each bar at room temperature, 400 F, 800 F, 1200 F, 1350 F, and 1500 F. The curves were made by applying a small pressure to the air column and waiting about 15 to 20 sec for the amplitude of the bar to become steady. Pressure and amplitude readings were then taken simultaneously. The pressure was then increased by a small amount, and the procedure repeated. By this process the total time of vibration of the bar might be as high as 10 to 15 min which may have a small effect on all subsequent tests. A family of curves from one bar is shown in Fig. 4, and the 1200 F curves from the three bars of the same material are shown in Fig. 5. These two sets of curves are shown only to indicate the nature and accuracy of the data; the composite curves to follow give a much better picture of the damping properties. It will be noted in Fig. 5 that the duplication of results is within approximately ± 25 per cent of the average of the three, and therefore any comparison between materials must show a 2 to 1 difference to be truly significant.

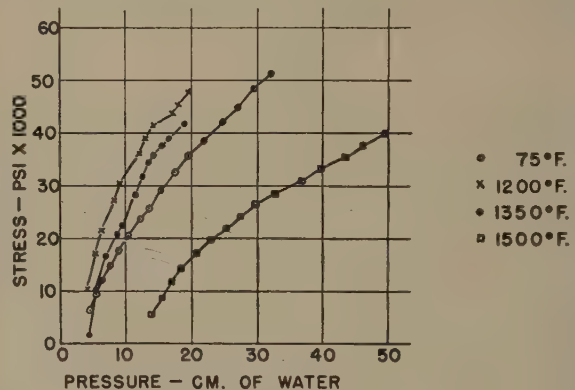

Fig. 4 Pressure Versus Stress for Inconel X at Various Temperatures

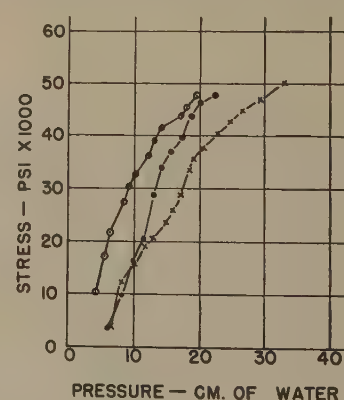

Fig. 5 Pressure Versus Stress for Inconel X at 1200 F on Three Different Bars

Experimental Results

The composite curves shown in Fig. 6 indicate the variation in damping at 40,000 psi with respect to temperature for the four alloys tested. Each point is the average of three bars. Note the minimum damping at 800 F for all but the iron-base alloys, as well as the extremely high values for N-155 above room temperature. The damping of N-155 was so high above 800 F that the pressures to produce 40,000 psi were above the limit of the gage

then available. Fig. 7 shows a similar series of pressure versus stress curves for S-816 at 1200 F, demonstrating an apparent temporary ceiling on stress. Curve A is the original curve made on increasing stress, and it will be noted that 80 cm of water pressure produced only about 47,000 psi (point 1). After about 10 min at 80 cm of water, the stress had increased to 54,000 psi (point 2). The automatic control was then turned on and the stress held constant for 2 hr, during which time the required pressure dropped to 40 cm of water (point 3). Curve B was then made while decreasing the stress, and curve C immediately afterward on increasing stress. Note the difference in curve shape probably resulting from not more than a few minutes' rest. Curve D, on the other hand, was taken on increasing stress after 15 hr rest at temperature. It is quite similar to the original curve (curve A) at low stresses, indicating a recoverable change

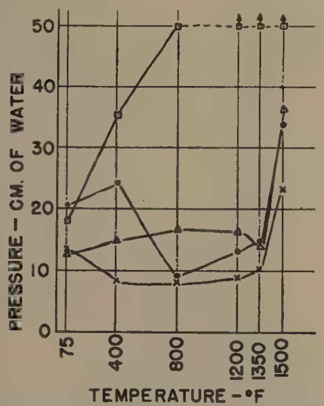

Fig. 6 Pressure at 40,000 Psi Versus Temperature

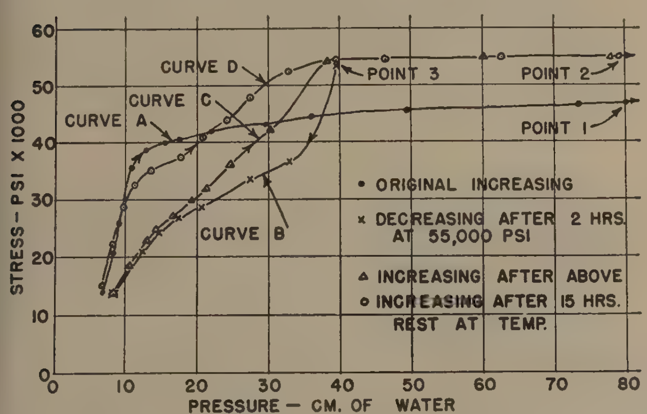

Fig. 7 Pressure Versus Stress for S-816 at 1200 F

in damping. This same type of damping change had been noted previously on 1020 steel at room temperature.[3] At high stresses, however, the curve rises to the "ceiling" produced by the previous 2-hr run which is considerably higher than the original ceiling.

Although not shown in Fig. 6, this process was repeated three times before the bar failed. Each time the low-stress damping dropped to its original value, and the ceiling was pushed progressively upward regardless of rest periods. The fatigue strength at 10^8 cycles of this bar was about 60,000 psi which means that all of the data in Fig. 7 were taken well below what is normally considered the damage point. The low-stress damping recovery is similar to the anelastic effect found in creep tests where some of the total creep is recoverable with time at a lower or zero stress.

After the tests of the type shown in Fig. 4 had been completed, one bar of each material was held at 1500 F and vibrated at 40,000 psi for approximately 50 $\times$ 10^6 cycles, or until failure. The second bar of each material was cooled to 1350 F and vibrated at 40,000 psi, while the third bar was cooled to 1200 F and vibrated at 60,000 psi. In all cases the pressure was recorded periodically. Figs. 8, 9, and 10 show the variation of pressure with time.

Referring to the data shown in Fig. 8, for 1200 F, after 1.5 hr of vibration under full power, N-155 had built up only sufficient amplitude to give 45,600 psi (60,000 psi was intended); at this point the control was turned on and the bar run at that stress. After only a few hours, the required pressure had dropped to a point below the other materials. S-816 showed the same characteristics, except that the intended 60,000 psi was developed. Had these materials been rated according to their initial damping,

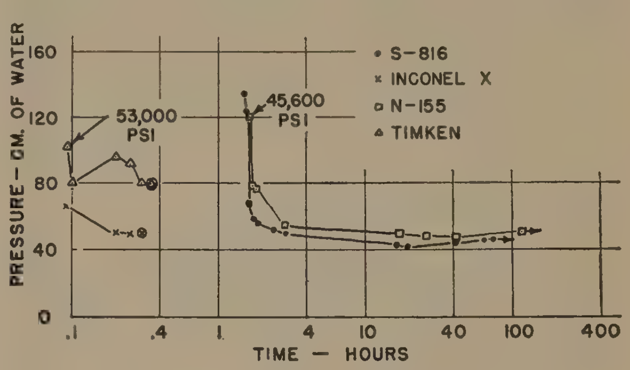

Fig. 8 Pressure Versus Time at 60,000 Psi 1200 F

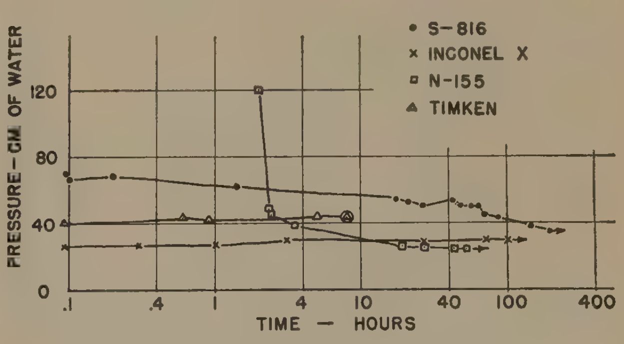

Fig. 9 Pressure Versus Time at 40,000 Psi 1350 F

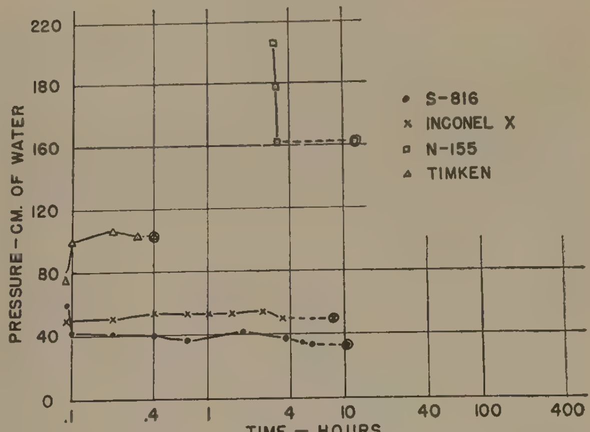

Fig. 10 Pressure Versus Time at 40,000 Psi 1500 F

[3] Technical Report No. 2, "The Damping, Elasticity, and Fatigue Properties of Materials and Structures Under Sustained Cyclic Stress." Sponsored by Office of Naval Research contract No. N6-ori-221; written by Syracuse University.

N-155 and S-816 would have topped the list by a wide margin, and yet after several hours of vibration, they shifted to the bottom of the list.

Fig. 9 shows the results of similar tests at 1350 F. It will be noted that N-155 retains its characteristic high initial damping and rapid drop. S-816 shows a much less rapid drop, being in the order of 2 to 1 in 100 hr with no evidence of becoming flat. Fig. 10 begins to show the same trends at 1500 F, but because of the more or less early failure of the bars, long-time results were not available. The points enclosed in large circles indicate the time of failure of the bar but not the pressure at failure. Those points, which are off scale to the left, indicate the initial pressure and should be plotted approximately one log-cycle further to the left (0.01 hr).

Summary

Tests on four gas-turbine-bucket materials have been conducted at elevated temperatures which involved the measurement of the pressure required to vibrate specimens in a pneumatically driven fatigue machine. The pressure is assumed to be a qualitative measure of the internal damping of the material.

It was found that at a peak stress of 40,000 psi, the nickel and cobalt-base alloys showed a minimum damping near 800 F and a sharp rise between 1350 F and 1500 F. The two iron-base alloys had a maximum damping between 800 F and 1200 F, a slight dip at 1350 F, and a sharp rise at 1500 F.

Two of the materials indicated excessively high damping at 1200 F and above at stresses below their fatigue limit. This high value, however, became rapidly smaller as vibration was continued, to the extent that they finally showed the lowest values of the four materials. This means that damping data obtained by more orthodox means in short times may, in some cases, be greatly in error if used in choosing materials for gas-turbine buckets. It appears that this could be a rather important factor in those designs where internal damping of the buckets is believed to be a major factor in limiting vibration.

In closing, it should be emphasized that these results are based on a minimum of data and are therefore subject to some revision as more data are accumulated. Probably the largest single source of error is the fact that a number of successive tests were made on a single bar, and the results themselves indicate that past history may have a marked effect on behavior.

Acknowledgment

The author is grateful to the Air Force Office of Public Relations for approval of this paper for publication.

Discussion

R. O. Fehr.[4] The author reports that the damping of materials varies with temperature, and shows a minimum value at 800 F. Tests previously conducted and reported[5] by us have shown similar results. Those investigations showed, on some materials, an increase in damping with temperature increase above room temperature. Further increase of temperature led to a reduction in damping. For example, the lowest damping was found for 13 per cent Cr-iron at approximately 900 F, for N 155 at approximately 1050 F, and for S 495 at approximately 1050 F, all for a stress of 30,000 psi.

[4] General Engineering and Consulting Laboratory, General Electric Company, Schenectady, N. Y.

[5] "Measurement of the Damping of Engineering Materials During Flexural Vibration at Elevated Temperatures," by Carl Schabtach and R. O. Fehr, *Journal of Applied Mechanics*, Trans. ASME, vol. 66, 1944, p. A-86.

H. F. Moore.[6] The author has presented an interesting summary of some preliminary results of damping tests of several high-temperature-resisting metals under repeated cycles of stress. While, as the author freely states, these results are not yet sufficient in number, or range of tests, to serve as a basis for quantitative values, the author is to be thanked for giving publicity to preliminary results in this field about which so much is yet to be learned. His use of a fatigue machine driven by compressed air in studying damping effects offers promise of usefulness in the study of change of damping, and his use of the variation of pressure as the tests proceed is worthy of critical study.

To this discusser the further detailed study of the phenomenon of damping seems to be another approach to the search for an experimental test which may possibly serve to locate the very early stages of the spreading fatigue crack, and thus to serve to shorten the long time necessary to find limiting stresses for oft stressed machine parts. It would seem worth trying to study the change of wave form of the damping curve after various lengths of time and different numbers of cycles of vibration, as well as to note the change of pressure.

In the investigation of the effect of repeated stress at high temperatures, it may well be remembered that in such tests creep as well as repeated stress affects the results, and the study of creep occurring in any tests, especially tests which take a long time to carry out, may yield some valuable information. Again the author is to be thanked for giving us a somewhat unusual viewpoint from which to study the behavior of metals under internal friction, especially metals at high temperatures.

Author's Closure

Work done since this paper was submitted and still continuing gives further support to the results reported. Comparisons now being made between damping curves and pressure values indicate a fairly linear relationship between pressure divided by stress and the logarithmic decrement. However, as expected, the machine damping is fairly high, being about 0.7 per cent at low stresses, down to 0.4 per cent at high stresses. By subtracting the machine damping quite reasonable values of damping have been obtained in the few cases where a direct comparison has been made to more orthodox damping tests. Due to the high machine damping it is obvious that very low internal-damping values cannot be accurately obtained.

The author wishes to thank the discussers for their consideration. The suggestions of Mr. Moore are being given careful thought and appear worthy of incorporation in our program.

[6] Research Professor Emeritus of Engineering Materials, University of Illinois, Urbana, Ill.

The Mechanical Seal—Its Construction, Application, and Utility

By C. E. SCHMITZ,[1] CHICAGO, ILL.

This paper outlines important construction features of the mechanical seal; discusses speed, pressure, temperature, and viscosity; application considerations, advantages, and when special construction is desirable; utility of the mechanical seal; and suggested services.

As a rather general statement, all mechanical seals basically are composed of a rotating and a stationary element, and some means of keeping them in contact (see Fig. 1).

As the number of and kinds of seals and their utility are appreciated by mechanical engineers, the number of papers and articles on the subject are increasing. The need for a better understanding of construction, application, and utility of this important mechanical device is becoming keener, and this paper is especially directed toward those important factors.

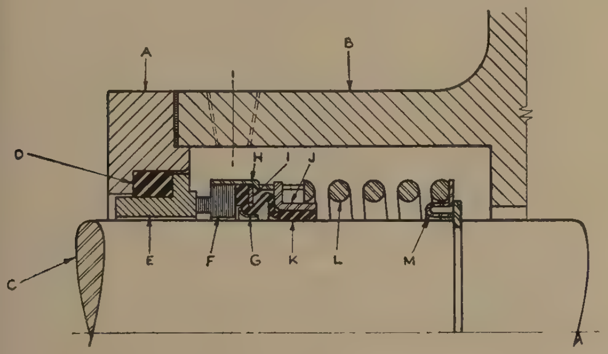

Fig. 1 Typical Mechanical Seal Applied to a Conventional Stuffing Box or Other Mechanical Device

(*a*, Gland plate or cover; *b*, stuffing box; *c*, shaft; *d*, floating seat ring, *e*, floating seat; *f*, sealing washer; *g*, shaft protecting ferrule; *h*, retainer shell; *i*, bellows flange retainer; *j*, driving band; *k*, synthetic rubber bellows; *l*, spring; *m*, spring retainer.)

Construction

Of primary consideration is the vital matter of construction, and here are involved the following important factors: (1) Flexibility; (2) surface finish of the mating faces; (3) positive driving means; (4) metallurgy; (5) engineering, laboratory testing, and research. These factors are not necessarily arranged in their order of importance. Each is important; all of them when properly co-ordinated make the most successful seal for the widest range of applications.

Since each mechanical device to which a seal is applied has manufacturing tolerances, the seal itself must have manufacturing tolerances. Therefore it will become immediately evident that flexibility is an important consideration. It is impossible to hold manufacturing axial tolerances below plus or minus 0.005, and this even in so-called precision equipment. Many high-production items run as much as plus or minus 0.020. Thus it becomes obvious that when initially installed, a good mechanical seal must be able to compensate for this axial tolerance, to which must be added the over-all length tolerances of the seal manufacturer, which on high-production items it is impractical to hold closer than plus or minus 0.015. It is evident that one might run into a condition of having to meet a total axial tolerance of plus or minus 0.035, and this is a conservative figure since, within the experience of many engineers, conditions of greater severity have been met (see Fig. 2).

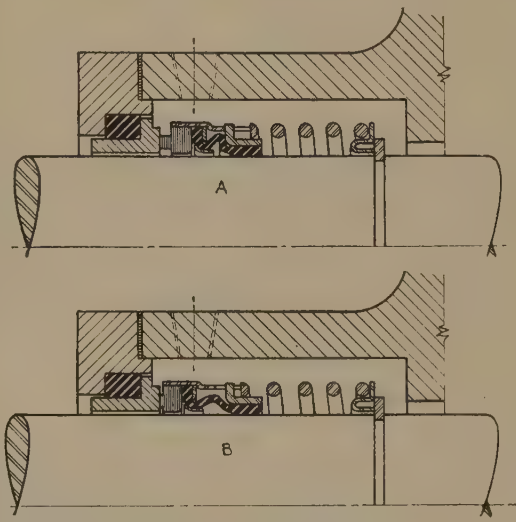

Fig. 2 (*a*), Mechanical Seal in Position for Use; (*b*), Axial Movement Desirable in Good Mechanical Seal Construction

Another factor wherein flexibility is important relates to radial tolerances which frequently range from plus or minus 0.002, almost the minimum, to as high as plus or minus 0.005. Again, these are reasonably conservative figures. The properly constructed mechanical seal, once set up in the installation, somehow must possess sufficient flexibility to compensate for wear at the contacting faces. Since wear is somewhat unpredictable, varying under several conditions of service, it is apparent that perhaps of highest importance in good mechanical-seal construction is flexibility. To insure the maximum in life, provision for not less than 0.090 wear should be made, since, when that much wear occurs, the seal has lost its utility and should be replaced.

As an example of just what a good seal might be called upon to do, a manufacturer, contemplating the use of mechanical seals, made an experimental hookup, and, as the shaft was rotated, struck it with a hammer and threw out any seal that leaked under this test. The radial movement of the shaft when the hammer blow fell was in the neighborhood of 0.035. Needless to say, many seal designers and manufacturers had their products thrown out. This may seem a hard and unjustifiable

[1] Vice-President and Director of Engineering, Crane Packing Company. Mem. ASME.

Contributed by the Machine Design Division and presented at the Annual Meeting, New York, N. Y., November 29–December 3, 1948, of The American Society of Mechanical Engineers. This paper was also presented at a dinner meeting of the Ontario Section of the Society on March 11, 1948.

test, yet in many mechanisms the shock and work-load conditions are such as to impose axial movement of considerable consequence.

A spring or series of springs is usually employed to keep the seal faces in contact as required to compensate for axial or initial tolerance. High loading usually will result in heating and excessive wear of the seal faces. The flexing members, therefore, must be so designed as to require low pressure to assure proper movement under installation and operating conditions. Materials used must be such that they will not work-harden and crystallize as so often occurs with thin metallic structures.

Specially compounded synthetic rubbers provide about the best flexing members. While other materials have been used, frequently they are discarded as not too satisfactory in so far as life and general good service are concerned. In addition to having a flexing member which will withstand considerable axial movement with the least possible effort, it is further necessary that the component parts of a good seal must be so designed as to permit the flexing member to move radially and axially freely, and under all conditions of operation.

The seat, usually stationary, should be so mounted in flexible materials that tightening of bolts or glands can in no way distort the closely lapped faces. Relatively heavy sections of iron or steel may be distorted by the pressure of one's hand so as to throw a closely lapped face considerably off flatness requirements and good seal practice. Some mechanical devices still are designed to use so-called solid end plates, but these are rapidly being discarded in favor of the floating seat. Floating, as it does, entirely in a synthetic resilient ring imposes no detrimental stresses beyond the flatness requirement and further permits ready replacement at minimum cost. Such seats are readily adaptable to most mechanical devices, and permit the seal user to purchase the whole unit from one source. A progressive seal manufacturer now supplies a complete seal unit, including a factory lapped and inspected stationary seat and rotating-seal washer, all having the required flexibility to meet the wide variety of operating conditions encountered.

Surface Finish of Mating Faces

Much has been written and said about surface finishes, primarily, perhaps, in regard to ordinary bearings where conditions are somewhat easier than encountered with seal faces. In one instance the rolling action of the shaft in the journal tends to roll in and on the lubricant, whereas seal faces usually are set in a vertical plane and present a somewhat more difficult problem in introducing proper lubricating films. Before the art of lapping and obtaining highly polished and parallel faces was commercially feasible, seal manufacturers used to ship seals and include with them a container of abrasive with instructions to set up the seal and put the abrasive paste between the faces, and thus wear them in. In rare instances some seal manufacturers follow this practice even today, but such practice is inconsistent with the trend and development of precision-lapping equipment.

The manufacturer recommending the abrasive paste between the seal faces never told the user how to remove the abrasive material from between the seal faces once its mission was accomplished, nor was any explanation given as to the effect of this abrasive material entering the oil or product handled. High surface finish eliminates the old so-called "break-in" period, permits more efficient and effective lubrication, by developing conditions which will eliminate oil-film failure by rupture through surface-to-surface contact. Costly experimentation and much research were necessary to work out the problem of equipment for producing at high speed satisfactory surfaces on the mating face of the seat and washer.

Nor was the selection of equipment all of the problem. Technique of handling and conditioning of lapping fluids, stress relieving and product handling, were also to be worked out to make the process adaptable to high-speed-production requirements.

There are two main objectives in the production of a good mating or bearing surface: The one, "metallurgical," is to remove the defective metal previously produced at the surface by the shaping or dimensioning operations and to expose the true crystals of the material actually bisected so as to have an extremely fine plane surface. The other, "geometrical," is to remove the hills and valleys (scratches and/or flaws) and, by the laws of physics, to generate a true and smooth surface which will eliminate the danger of film rupture and material-to-material contact which lead only to increased friction wear and failure.

The following test data compiled at the Massachusetts Institute of Technology may be of interest to mechanical engineers: This test was run to demonstrate the failure point of load-carrying capacity of the oil in a conventional bearing. The oil used had a viscosity of 150 sec at 110 F, and was grade MS 782. The shaft speed was 950 rpm. With a finish of from 22 to 28 microinches, the failure point was an average of 71 lb. With a finish of from 8 to 10 microinches, the failure point was 153 lb average. With a finish of from 0 to 2 microinches, the failure point was 217 lb average. While these figures may not be wholly applicable to vertically running faces, they are at least indicative of what takes place with regard to seal faces.

It was early discovered that root-mean-square (rms) specification, while important, was not the only problem to be considered. Flatness of the whole surface area is also important, and to this end optical flat inspection is required. Many scientific and engineering organizations are doing work on surface finishes. Among those interested and working with the problem are the National Bureau of Standards, the Society of Automotive Engineers, and the National Aircraft Standards Committee.

Equipment to evaluate surface conditions is being improved and made available to those interested in precision manufacturing. Principal among the manufacturers of such equipment are the Physicists Research Company, Brush Development Company, Compar Company, and the several manufacturers of optical flats.

The progressive seal manufacturer inspects, with optical flats, all lapped surfaces which he produces; spot checks will not suffice. One hundred per cent inspection of each and every part that is manufactured is necessary to assure users the best possible precision faces, optically flat, smooth, and thus designed to give the maximum of efficient long life. Lately some experimental work has been done with so-called liquid honing whereby abrasives ranging from 400 to 600 mesh are forced against the sealing faces at extremely high velocities. It is yet too early to predict whether or not this surface treatment will be beneficial in so far as seal construction and production are concerned.

Positive Driving Means

A means of so driving the seal that none of the torque load can be or is imposed on the flexing member, whether it be of metallic or synthetic materials, is important (see Fig. 3). The washer or usually moving face may require considerable breakout torque and, if the flexing member is called upon to carry any part of this load or the running load, it soon crystallizes, especially if the membrane is metallic. If it is of synthetic rubber, internal abrasion and constant flexing cause the material to break down rapidly, and seal failure of course results.

Good practice requires that the flexing member must be so constructed as to permit close contact of the faces to compensate for wear. High unit loading tends to exclude the thin lubricating

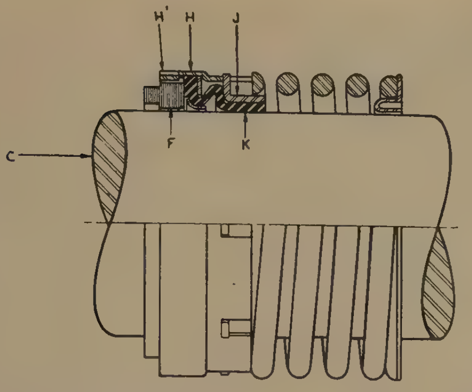

FIG. 3 SYNTHETIC RUBBER BELLOWS (k) ADHERES TO SHAFT (c) ND DRIVES DRIVING BAND (j) WHICH DRIVES RETAINER SHELL (h). DRIVING NOTCHES h' IN RETAINER DRIVES WASHER (f)

ilm. The manufacturers of many seals on the market today ttempt to use high spring pressure and thus impose the driving-orque load on the spring. Frequently this results in chattering, oor seal life, and general dissatisfaction.

METALLURGY

Good general metallurgy necessitates selection of proper ma-erials to withstand the medium in which the seal operates. This problem is not too severe in most industrial equipment, vhere brass and steel are most commonly used for sealing metal arts. These serve well in most oils, waters, greases, and hydro-arbon distillates. Seals are finding an ever-increasing use in the ighly complex chemical industries where corrosion is a common roblem. In these instances, stainless steel, is commonly used. Iowever, in considering seal metallurgy, of highest importance re the washer and seat-material combinations. For most indus-rial installations combinations of a good grade of cast-iron seat vith carbon or synthetic-resin-bonded graphite-and-metal wash-rs make for long, satisfactory life. Such combinations are videly used in refrigeration service and are well-suited for rela-ively high-speed industrial applications, where the principal roblem is to hold a lubricating medium, and perhaps gases such s these commonly employed in the refrigeration field.

Carbon and bronze serve well for problems involving water. Meehanite and "ni-resist" run well with carbon combinations in ertain types of installations. In highly specialized services tellite, steel, ceramics, and a host of other materials are run various combinations which testing and field service dem-nstrate are best-suited. As the need for and use of seals ontinue to increase, new combinations of materials are being ried. Almost every known combination of metal, carbon, and eramic product are experimented with and then used in actual nstallations after research and laboratory testing reveal that iere is a possibility of such materials being employed to the dvantage of the seal user.

ENGINEERING, LABORATORY TESTING, AND RESEARCH

A seal can only be as good as the engineering, testing, and esearch which go into it. Engineering involves not only applica-ons but structural designs as well. No reference work being vailable, all developments must go through extensive laboratory sting to prove or disprove their utility. As an example, the atter of mating-face widths: A start is made with what seems ieoretically practical, followed by actually testing various com-nations of wider and narrower faces. Each one, after the iginal idea, must be tried until at last the most desirable is lected for operation in a given material at selected speed, temperature, and pressure. Handling another material may so change the results that another complete set of tests is necessary.

Testing should not be confined to any one particular type of equipment. A good variety of pumps and mechanical appliances should be available so that tests can be conducted over a rather wide range of services, and thus establish the most practical seal construction for the specific problem. There should be available in the mechanical testing laboratory means of testing under a wide variety of speed, pressure, and temperature conditions. Each test should be recorded carefully and the seal so tested should be catalouged carefully and kept available for observation at some later date should it be necessary.

The engineering organization should be comprised of those who are especially trained in making applications to a wide variety of mechanical problems. It should include specialists in application principles, and others who are specialists in the art of design for development of the various parts which go to comprise the seal itself.

Supporting the engineering department and the mechanical testing laboratory should be available the facilities of a completely equipped research laboratory to analyze various materials to be handled and to assist in selecting construction materials best-suited for any given problem. All this assures the user of mechanical seals the best equipment that money can buy.

SPEED

Speed is of interest to the prospective seal user, especially in some of the newer mechanical appliances where speeds of 10,000, 15,000, and 20,000 rpm are being considered rather generally. Tests so far conducted seem to indicate that about 12,000 rpm or 5,000 fpm (whichever is the lower) at the seal faces is about the maximum that can be tolerated. The mechanical appliance to which the seal is to be applied, and the liquid to be sealed are factors which materially influence maximum speeds permissible.

The problem of excessive speed is not generally encountered in centrifugal pumps where speeds of 3600 to 4000 rpm are most generally used. In positive-displacement pumps speeds rarely exceed 1200 rpm, and therefore speed, in so far as the mechanical seal is concerned, is not too important in this type of equipment. Centrifugal blowers, having shaft speeds of from 6000 to 9000 rpm, present a real problem for the seal designer, owing to the large shaft sizes usually involved. Progress is being made and each new requirement leads to new research and study.

Spring behavior must be studied when considering speed limits, since experience has shown that springs behave rather unexpectedly under the influence of centrifugal force. The stroboscope proves a handy instrument in this investigation. By running loaded springs at various speeds, the behavior of any particular spring or series of springs can be observed and designs can be arranged to suit the required condition.

RECOMMENDATIONS

Seat and Washer Material. Since there are few data available concerning the best-suited washer and seat materials, and since these parts running together do not follow the pattern of the sleeve bearing, elaborate tests should be set up and run to determine the most suitable materials for any given set of conditions. Carbon products which include various grades that are filled with metals, carbon, or some synthetic substance, or synthetically bonded carbon, asbestos, metal combinations, make for less friction and show good wearing qualities under general service conditions, and are widely used.

Carbon washers running against close-grained cast iron, can be used successfully for jobs involving various grades of lubricating oils, distilled hydrocarbons, soluble oil, and similar

products. When handling hydrocarbons having a specific gravity of 0.55 or under, stellite shows excellent results. "Ni-resist" and carbon combinations do well in clear or gritty water, sea water, all grades of lubricating oil, and petroleum distillates, such as gasoline, kerosene, naphtha, and others having relatively high hygroscopic characteristics.

Viscosity. It has been determined that viscosity plays an important part in seal construction. Tests indicate that up to 3000 sec SUV, standard seal construction can be used. Over 3000 sec SUV, special study and experimental work are necessary. Work thus far indicates that, as viscosity increases, spring pressures should be increased to prevent excessive increase in film thickness, since if the film thickness increases then ability to hold the liquid decreases. The major number of seal installations, falling as they do in the viscosity range of 3000 sec SUV and under, no exhaustive study has yet been undertaken. It is hoped that in the not too distant future this subject can be studied and a report of the findings issued at that time.

Dry Running. So-called dry running must be avoided in mechanical-seal usage, since the materials used for seat and washer tend to distort or abrade at temperatures of over 400 F in most instances (see Fig. 4).

It is recommended that provision be made to insure liquid in the seal chamber at all times. Usually, this can be accomplished in the case of pumps, by providing a line from the pump discharge to the seal chamber. In mechanisms other than pumps, the liquid sealed usually can be brought in constant contact with the seal faces.

No combination of materials has yet been discovered which can tolerate dry running for more than a few minutes' duration. This situation must be watched carefully when handling materials which lose liquid phase under certain conditions of temperature and pressure. When handling extremely volatile liquids, pressure in the seal chamber must be maintained sufficiently high to insure liquid phase at all times.

Pressure. The properly designed mechanical seal can be used for a wide range of pressure conditions. Standard seals can be used for vacuum service down to 0.5 mm of Hg abs, and on pressures up to 200 psig without difficulty and with any liquid in which a seal can be operated. Mechanical seals were used the 2500-ton cyclotron at Columbia University where vacu down to 10^{-5} mm of Hg had to be maintained constantly, marily to give the swiftly moving ions a free path of moveme When pressures of over 200 psig are encountered in the s chamber, balanced seal construction is recommended (see Fig.

The balanced seal is so constructed that the seat and was faces are in pressure balance. The hydraulic pressure within seal cavity is so disposed as to balance itself and thus only usual spring loading is necessary to compensate for wear. Lon life, less friction, lower heat generation, and minimizing pov requirements are the result of balanced seal usage. The n chanical seal is not handicapped by pressure shock or pulsati Pressures up to 1000 psig can be handled regularly and safe Higher pressures can be handled, but here again special design a construction are required.

Application. In considering the problem of applying

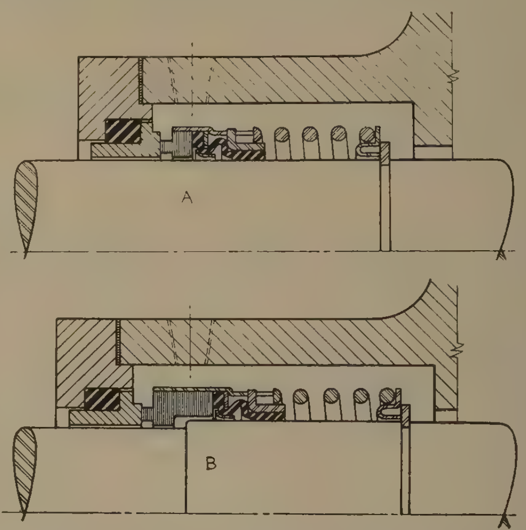

Fig. 5 (*a*) Typical Standard Mechanical Seal Constructi (*b*) Typical Balanced Mechanical Seal Construction

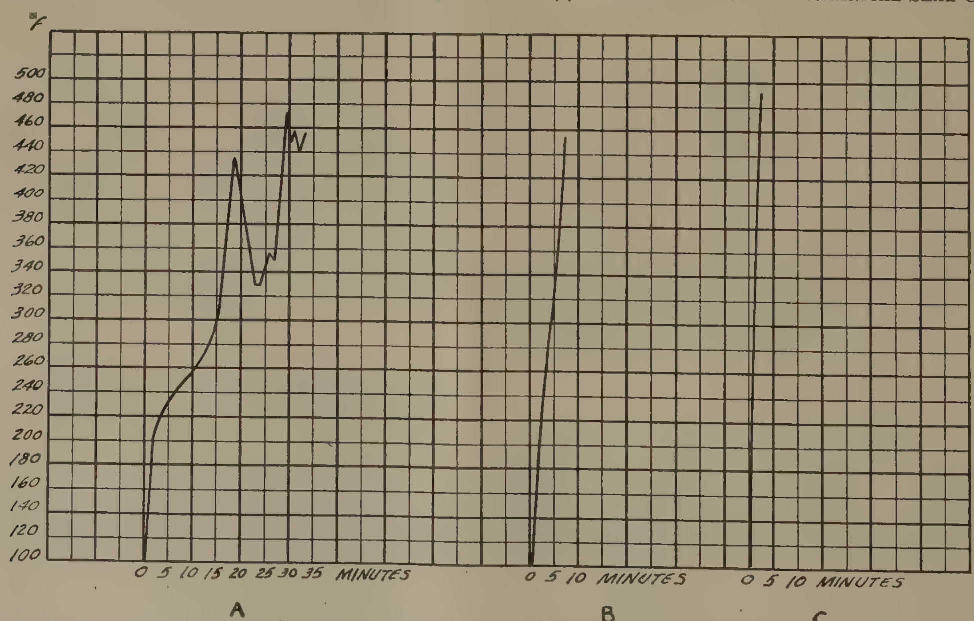

Fig. 4 (*a*) Time Interval and Temperature Rise When Seal Faces First Run 24 Hr With Liquid in Cavity, Then Drain and Run Dry. (*b*) Time Interval and Temperature Rise When Seal Faces Oiled Before Starting Run; No Oth Lubrication or Cooling Provided. (*c*) Time Interval and Temperature Rise When Seal Faces Installed and Run D All Above Tests at 3600 RPM

mechanical seal, the prospective user nearly always comes to the manufacturer with the design of his equipment already worked out. Space requirements are held to the very minimum, usually based on some preconceived idea of what ought to do the job. After initial tests are run, the seal selected for solution of the problem falls short of the needs, and a solution must be sought, usually hedged with this statement: "Anything used must fit into the already existing space." This is generally limited further and is too small in both axial and radial dimensional requirements for application of the proper mechanical seal.

Engineers having sealing problems would do well to submit the problem in the early design stages in order to avoid redesign and costly changes later. In a great many instances, considerable savings can be realized by embodying construction applicable and already widely used when mechanical seals are employed. Of special interest to mechanical engineers and the progressive equipment builder is the application of a combination of talents in the early development stages, since a properly sealed unit may be constructed to take advantage of the following:

1 A more expensive lubricant better-suited to the problem, employing seals to prevent loss of the lubricant.

2 A lubricant of lower viscosity, since it can be held satisfactorily by seals.

3 Elimination of costly filtering and accompanying equipment, since seals prevent contamination of oil in the system.

4 Freedom from fire and personnel hazards when handling either explosive or toxic materials.

5 Freedom from necessity for frequent adjustment of glands, which often is placed in the hands of inexperienced personnel.

6 Full utilization of input power for the equipment intended, since, frequently, packing glands with conventional packing, are drawn up so tightly as to consume a large percentage of the power input, and thus make it necessary for manufacturers to use larger prime movers than are necessary.

Applications have been worked out for hundreds of appliances, and layouts are available without cost on the part of the potential seal user. However, to obtain the most satisfactory performance, the mechanical seal must be selected and applied carefully. When so used such seals will give years of satisfactory service without attention or adjustment. A big factor in favor of using mechanical seals is that they eliminate the human element entirely. Once properly installed, no adjustment is required and they can be forgotten.

In making applications, care should be exercised to insure the pump material or lubricant contacting the seal faces. Early packing designs employed many ingenious devices to keep the lubricant or material sealed away from the gland. In applying seals, exactly the opposite practice should be employed. Higher oil levels should and can be maintained, and the equipment manufacturer who uses seals intelligently can point out to the customer the considerable savings which can be realized in maintenance and replacement of the lubricant.

The machine designer must keep in mind that shaft rigidity is important, and deflection should be kept to the minimum, since various changes take place in iron and steel parts if load is applied and taken off. If the load were steady, deflection would not present too great a problem, but it is the constant changing with operating conditions which complicates the matter. Proper alignment makes for better and longer equipment life. Its importance cannot be disregarded when applying mechanical seals.

Double seals are often applied advantageously, especially to overcome corrosion problems, or when materials, containing large percentages of solids, are handled (see Fig. 6). Some lubricating medium must be circulated between the double seals so as to provide proper lubrication for the seal faces, and to carry away such heat as is generated. These installations are widely used in paper mills, chemical plants, or when handling highly irritating gases. On such installations, the mechanical engineer is called upon to select the oil or liquid circulated through the seal cavity.

Single lubricated face seals are applied to jobs where the pressures are low and the material handled has poor lubricating qualities, or contains materials which might be detrimental to the highly polished faces (see Fig. 7). The lubricating material is introduced through the stationary seat, the seat having a series of holes leading from the face to the lubricating-material supply

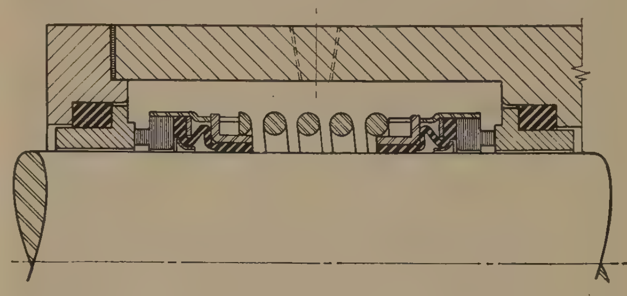

Fig. 6 Typical Double-Seal Construction. Lubricating or Cooling Liquid Is Pumped Through Stuffing Box at Pressure 10 to 20 PSI Greater Than Pump Discharge Pressure

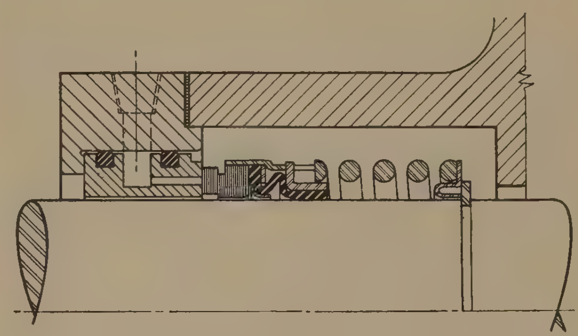

Fig. 7 Typical Lubricated Seat Construction

labyrinth. Lubrication is applied usually by means of a spring-loaded grease cup, using light, specially designed grease; or oil which is introduced by a circulating pump or from a gravity storage supply, whichever is indicated to be the best, based on laboratory and field tests for the particular problem at hand.

Utility of Mechanical Seals

The mechanical seal already has demonstrated its utility in many industries and on a wide variety of equipment. To recite a few, the refrigeration industry was perhaps the first to utilize the effectiveness of the mechanical seal, and manufacturers of large and small compressor units have used them for a considerable length of time. There is hardly a progressive refrigeration manufacturer of either large or small units who does not today make full use of the mechanical seal. Pump manufacturers everywhere are offering customers the advantage which can be obtained by their application on pumps for a large variety of services which include the following:

Domestic units for handling water in small and large well installations.

Paper mills for handling stocks and black liquor.

General service equipment, where pumps are called upon to handle brine, quench oil, soluble oil, and a host of applications which will present themselves immediately to the reader under this category.

Pumps on transcontinental pipe lines, where conventional mechanical packing meant tolerating rather excessive leakage which resulted in a serious problem of repumping that leakage into the pipe lines.

On pumps in petroleum refineries, where conditions present a host of problems, pump manufacturers today are recommending seals on every conceivable type of unit in the processing industry.

Speed-reducer manufacturers have long felt the need of freedom from gland leakage, and most progressive manufacturers are now using the mechanical seal.

Manufacturers of agitators and vacuum-processing equipment find the mechanical seal a solution to many of the problems which have troubled them previously.

The use of seals on the coolant pumps on internal-combustion engines is an important development. Today, with but one or two exceptions, every automobile builder uses seals to the comfort of many automobile owners.

The Diesel-engine manufacturers are making wide and increasing use of seals on coolant and oil-transfer pumps, reducing "in-shop" time and making for cleaner and more efficient operation. Frequently the adoption of mechanical seals results in considerable construction savings, eliminating parts and expensive gland construction. Mechanical seals permit full streamlining and enable important space-saving requirements to be made.

Summary

Construction is important, and every interested prospective user should acquaint himself fully with the desirable factors and necessary requirements for his particular job and insist that the seal used possess all of the desirable factors that go to make up a good seal.

Application of the proper seal to the job is the answer to successful performance. Submit application problems to a qualified manufacturer before they are too far advanced in construction details. Follow closely the layout submitted and approved for the particular job under consideration, and thus assure a successful seal application.

Utility is unlimited, and the prospective seal user would do well to deal only with a manufacturer competent and capable of analyzing the specific problem and willing to work with the equipment manufacturer toward a successful solution of the problem involved.

Discussion

C. L. Pope.[2] The author refers to the ability of the mechanical seal to prevent leakage. It is our opinion that a mechanical seal must permit leakage if it is to run as a lubricated surface. If the seal is not lubricated, wear will result. It is granted that under the right conditions wear is low and relatively long seal life can obtained. Seals should last indefinitely if a seal controls leakage so that the surfaces are at all times lubricated.

P. C. Stein.[3] Exception must be taken to the inference th dry-running of mechanical seals is not possible. Dry seals ha been run and are being run on commercial machines at rubbi speeds far in excess of the limits which the author gives for liqu seals. The author states: "No combination of materials h yet been discovered that can tolerate dry rubbing for more than few minutes' duration." Certainly, he is aware that motor-a generator brushes have been able to tolerate weeks of continuo operation with dry-rubbing contact since well over a half centu ago. This example goes somewhat afield from mechanical seali devices, but it is one which is commonplace in everybody's e perience. Without question, dry-rubbing contact per se at su stantial speeds is possible.

The writer has designed and has been associated with the manu facture and testing of well over a hundred dry seals which ha given satisfactory service in the field, some at high rubbing spee Several typical applications are given in Table 1.

Leakage measurements were not made on the applications giv in Table 1 beyond determining that seal performance was sati factory from an operational standpoint, with the exception applications Nos. 2 and 3. In application no. 2, for which seals were furnished, leakage test results were below 4 cfm free a per seal at the operating pressure. Application no. 3, which w in a compressor for a supersonic wind tunnel, did not have direct leakage test for the seals after installation, but a test in cated that leakage into the enire tunnel, including leakage throu the compressor casing, valve stems, piping flanges, and gaskets the working-section entry ports under full vacuum, as well through the two seals, was less than 0.4 cfm of free air.

In general, leakage of dry seals can be held to almost any re sonable minimum by refinements in manufacture and workma ship. In most applications, the additional expense incurred obtain very low leakage is not justified. In application No. which is a shop experimental installation, it is consistently pos ble to have the leakage per seal less than 0.75 cfm of free air at 1 psig internal pressure. This installation has been operated speeds of 19,000 rpm with test durations extending over seve days.

Experimentally, the writer has operated a 10-in-diameter se intended for dry operation on a gas-turbine dummy piston speeds up to 8800 rpm (rubbing speed of 23,000 fpm). The te were being conducted in an effort to test seal performance 15,000 rpm (rubbing speed of 39,200 fpm), but bearing and co pling failures terminated the test on every attempt to attain t desired speed. Seal performance under 60 psig air pressure b fore mechanical failure of the testing machine was always good

[2] Lubrication Engineer, Eastman Kodak Company, Rochester, N. Y.

[3] Consulting Engineer, The Kuchler-Huhn Company, Inc., Shar Hill, Pa.

TABLE 1 APPLICATIONS OF DRY SEALS

Application no.	Seal diam, in.	Rpm	Rubbing speed, fpm	Type of machine	Medium sealed	Pressure sealed	Ambient temp, deg F
1	6 3/8	4000	6660	Compressor	Air	45 psig	260
2	6 7/8	4000	7180	Compressor	Air	75 psig	375
3	7 1/4	4660	8830	Compressor	Air	{ —27 in. Hg to +12 psig	...
4	6 3/8	8300	13820	Compressor	Air	{ —27 in. Hg to +12 psig	...
5	2 1/2	1800	1180	Autoclave	Air	250 psig	300
6	1 1/2	1800 and	706 to	Centrifuge	Volatile	Nominal	...
	5 5/8	3600	5290		Vapor		
7	4 5/8	1000	1210	Textile draw-roll	Steam	400 psig	448
8	3 1/2	19000	17400	Experimental	Air	100 psig	80

The speed limitations given by the author for liquid seals are far below what has been attained to date even in commercial service. The writer has designed and has been associated with the testing of liquid seals for high-pressure compressors which are in service with rubbing speeds well in excess of the author's limitations. He has also conducted tests of similar seals at rubbing speeds of roughly twice the upper limit given by the author.

In the present state of the art of seal design, it is rash to venture an opinion of the possible upper limits of rubbing speed for either dry, nonlubricated, or liquid-lubricated mechanical seals. The limitations of a given design are determinable, but recognition of the mechanism and properties which cause the limitations will often extend the regions of operability of a mechanical seal.

Author's Closure

Mr. Stein overlooks entirely the type of sealing that is being discussed in the author's paper. In the first instance, let us consider the rather thinly drawn analogy to motor and generator brushes. It has been the consensus of opinion of electrical engineers that there exists between the brush and the armature an air film which serves to prevent excessive friction, wear, and heat. The correctness of this air-film theory is rather well borne out by the fact that unlimited difficulties were experienced in high-altitude flying where the air film was of low degree, which resulted in rapid wear, excessive friction, and the generation of considerable undesirable heat.

The seals discussed in the author's paper are such that the loss of even 0.4 cfm would be unthinkable and intolerable. It can be appreciated readily that if leakage amounting to 0.4 cfm existed, it would be impossible to live in a room where an ammonia compressor was in operation. There are many other toxic problems where such losses could not be tolerated.

The seals primarily discussed and considered in the author's paper are those wherein the losses of oil in refrigeration compressors, for example, must be held well below 10 grams per 1000 hr of operation. The author does not deny the possibility of operating a seal on an air or gaseous film at relatively high speeds, provided losses such as Mr. Stein proposes are not objectionable.

In order to hold leakages to extremely low limits (which is understood to be of such a low order that the gas is not detectable by conventional means), the imposition of considerable spring loading is required. This, in the opinion of the author, makes it impossible to run dry. Innumerable tests of various designs and construction with innumerable material combinations have proved the impracticability of attempting to operate dry. It is the opinion of this author that high speeds with dry running are possible only when relatively high leakage can be tolerated.

If Mr. Stein will take into consideration the difference in sealability of the seals discussed in the writer's paper and those brought forth in his discussion, he should agree readily that the arguments he advances have little or no bearing on the precision type of low-leakage seals covered in the author's paper.

Cyclic Heating Test of Main Steam Piping Joints Between Ferritic and Austenitic Steels—Sewaren Generating Station

By H. WEISBERG,[1] NEWARK, N. J.

A cyclic heating test of several full-size heavy-wall pipe joints between austenitic and ferritic steels is described. It is concluded that sound welded joints can be made between these dissimilar materials and that such joints will withstand the effects of temperature changes which may be expected to occur in modern power-plant service.

Stainless-steel steam piping and valves are new in power-plant practice. While various austenitic steels have been used for several years in oil-refinery and other high-temperature industrial-process piping, these applications are at low pressures, requiring relatively thin-wall pipe. Also, the expected life is comparatively short. Sewaren Generating Station of the Public Service Electric and Gas Company, New Jersey, will be the first steam power plant to operate at 1050 F initial steam temperature, the steam pressure being 1500 psi. Both the General Electric and Westinghouse 100,000-kw turbines have been furnished with austenitic-steel valves and inlet steam piping. The Combustion Engineering Company boilers, however, are provided with 3 per cent chrome 1 per cent molybdenum (ferritic) steel piping from the superheaters to the turbines. This paper describes a test of several full-size joints between the austenitic and ferritic steels required for the piping of the Sewaren installation.

The connection between the boiler leads and the turbine stop valves is $12^3/_4$ in. diam and 2 in. thick. The suitability of welds between dissimilar materials in this heavy section was the subject of considerable discussion and speculation. During the past few years a number of replacements of stainless-steel piping systems and valves in oil-refinery service have been reported, due to cracking. These failures, however, all occurred in cyclic processes in which temperatures change rapidly at frequent intervals, a much more severe operating condition than may be expected in power-plant service. Also, welds between ferritic and austenitic steels have been reported to have developed cracks in service near the joint. These failures were in connection with certain mercury-boiler applications; a number of high-carbon-steel welded steam-boiler drums in Germany during the last war; and aviation gas-turbine wheels where austenitic blade rings were welded to ferritic wheels. It was reasoned that because the coefficient of thermal expansion of austenitic steel is about 50 per cent greater than that of ferritic steel, exceedingly high stresses were set up with changes in temperature which resulted in cracking at the joints. On the other hand, a number of cases in the same applications were reported to be satisfactory under equally severe operating conditions. It is entirely possible, therefore, that the failures were due to other causes.

Paralleling the design of the piping systems for the Sewaren units, it was decided to conduct a test of several full-size welds and subject them to 100 cycles of heating and cooling, simulating operating conditions. Single-boiler turbine-generator units on the interconnected Public Service System are designed to run continuously with scheduled shutdown once a year, and only occasional emergency shutdown for repairs. Alternate heating to 50 F above the operating temperature and cooling to the vicinity of the ambient temperature 100 times should give a good indication of conditions at the end of a major portion of the life of the unit. The long-time stability of the material at operating temperature of course would not be evaluated in the short-time test. Also, the effect of stresses due to internal pressure and external expansion forces present in the actual piping system would not be reflected. These stresses, however, were not considered sufficient to warrant the expense of inclusion in the test, as they are minor compared to the stress which is set up due to differential expansion of the dissimilar materials, and the stress due to temperature difference across the pipe wall when cooling rapidly. The latter condition may occur as a result of accidental water carry-over from the boiler. In order to simulate this condition, a series of ten quench cycles was included in the program following the "normal" heating and cooling tests.

Description of Test Pieces

In the summer of 1946, a co-operative arrangement for making the test was agreed upon among representatives of the Public Service Electric and Gas Company, the General Electric Company, the Westinghouse Electric Corporation, the Combustion Engineering Company, and the M. W. Kellogg Company. A length of 12.75-in-diam × 1.625-in-thick $2^1/_4$ per cent chrome 1 per cent molybdenum forged and bored pipe, available from the Essex installation, was cut into short spool pieces and welded to a series of 18/8 castings, using a number of austenitic filler rods of various compositions. The assembly, shown diagrammatically in Fig. 1, is hereinafter referred to as the multiweld

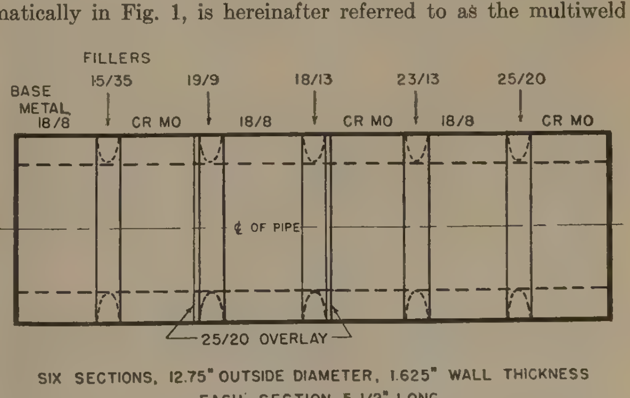

Fig. 1 Arrangement of Multiweld Test Piece

[1] Mechanical Engineer, Electric Engineering Department, Public Service Electric and Gas Company. Mem. ASME.

Contributed by the Power and Metals Engineering Divisions with the Joint Committee on Effect of Temperature on Metals and presented at the Annual Meeting, New York, N. Y., November 28–December 3, 1948, of The American Society of Mechanical Engineers.

TABLE 1 DETAILED INFORMATION ON MATERIALS IN MULTIWELD AND KELCALOY TEST PIECES

Multiweld Section:			
18/8 Stainless-steel rings—Type 347 castings (Midvale)			
C	= 0.09	Si	= 0.76
Mn	= 0.41	Ni	= 11.03
P	= Trace	Cr	= 17.64
S	= 0.011	Cb	= 1.01
Tensile, psi		71000	
Elastic limit, psi		36500	
Elongation, per cent		24.1	
Reduction of area, per cent		38.5	
$2^1/_4$ Cr-1Mo steel rings including backing strips, ASTM A-213 Gr T22, modified (Penn Forge)			
C	= 0.06	Si	= 0.29
Mn	= 0.51	Ni	= ..
P	= 0.015	Cr	= 2.24
S	= 0.013	Mo	= 1.0
Tensile, psi		63750 to 69300	
Yield, psi		29250 to 43650	
Elongation, per cent		29.5–35	
Reduction of area, per cent		63–69.5	
Brinnell		163 max	
2 Kelcaloy Section:			
Outer casting (M. W. Kellogg)			
C	= 0.151	Si	= 0.53
Mn	= 0.11	Ni	= 11.86
P	= 0.016	Cr	= 19.16
S	= 0.01	Cb	= 1.125
Mandrel, $2^1/_4$ Cr-1Mo, forged steel (Penn Forge)			
C	= 0.08	Si	= 0.29
Mn	= 0.43	Ni	= ..
P	= 0.020	Cr	= 2.19
S	= 0.013	Mo	= 0.98
Tensile, psi		71000	
Yield, psi		33000	
Elongation, per cent		34.5	
Brinnell		163 max	

3 Filler Metals:

Multiweld section—lime-coated 5/32 in. diam (Arcos Corp.)

Type	Analysis
Chromend 15/35	Cr 14.70–14.79 Ni 33.55
Chromend 19/9	Cr 18.98 Ni 10.18 Cb 0.97
Chromend 18/13	Cr 17.37–17.41 Ni 11.74 Cb 0.70 Mo 1.89– 1.91
Chromend 25/12	Cr 21.92–22.00 Ni 12.76 Cb 0.84
Chromend 25/20 (and overlay)	Cr 24.45 Ni 20.73 Cb 1.1

piece. More detailed information on the materials is given in Table 1. All welding was done with a preheat temperature of 600 F, and the piece was subjected to a postheat of 1550 F for 3 hr followed by furnace cooling. Fig. 2 shows the welded piece before machining. Fig. 3 shows the same piece after machining and removal of trepanned coupons which served as a record of the condition of the welds before the heating tests. The contrast between the austenitic and ferritic steels can be seen clearly in the latter illustration.

At the suggestion of Mr. D. B. Rossheim of the M. W. Kellogg Company, an additional joint of an entirely new type was fabricated, as shown in Fig. 4. The purpose of this arrangement is that the major portion of the interface between the dissimilar metals is longitudinal to the pipe rather than transverse, and therefore subject principally to shear stress only as a result of internal pressure or bending forces. If failure occurs at the joint between the two metals, it should appear as a lap or lamination rather than as a separation straight across the pipe. The joint was made by the Kellogg "Kelcaloy" process, which is used in pressure-vessel construction where low-alloy base metals are clad with corrosion-resistant materials.

Fig. 5 is an external view of the transition piece containing the joint, hereafter referred to as the Kelcaloy piece. The compositions of materials used are given in Table 1. A short piece of $2^1/_4$ per cent chrome 1 per cent molybdenum pipe was welded to the ferritic end of the test piece in order to provide a weld between similar materials for control purposes.

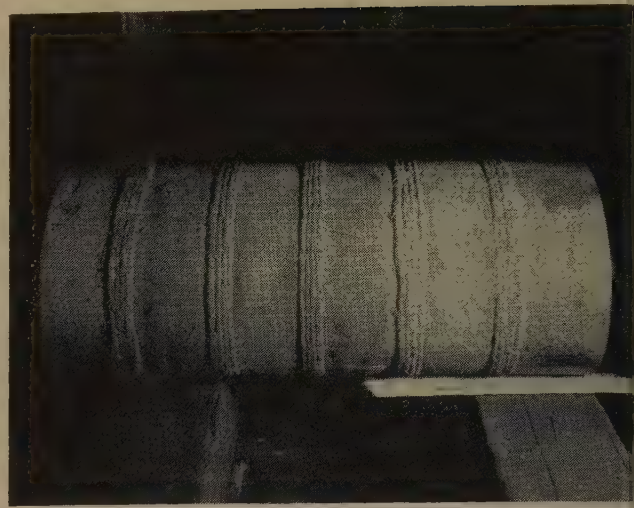

Fig. 2 Multiweld Test Piece Before Machining

Fig. 3 Multiweld Test Piece After Machining and Trepanning of Test Plugs

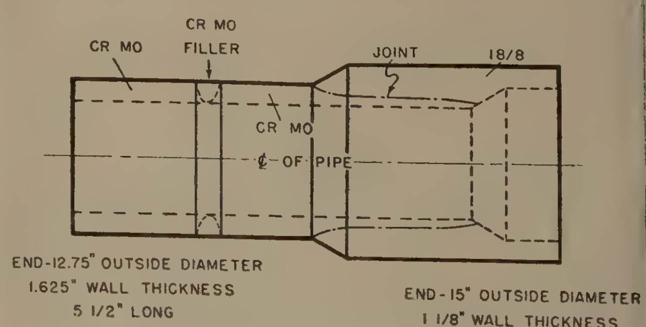

Fig. 4 Arrangement of Kelcaloy Test Piece

Fig. 5 Kelcaloy Test Piece After Machining and Trepanning of Test Plugs

Heating-Test Procedure

The heating of the test pieces was carried out at the Essex Station of the author's company, where induction-heating equipment was available. Initially, it was proposed to heat the test pieces directly by winding them with induction coils. It was found, however, that it was impossible to heat uniformly owing to the widely different magnetic properties of the two steels. The final arrangement was a furnace which consisted of a 24-in. insulated induction-heated pipe, in which both test pieces were

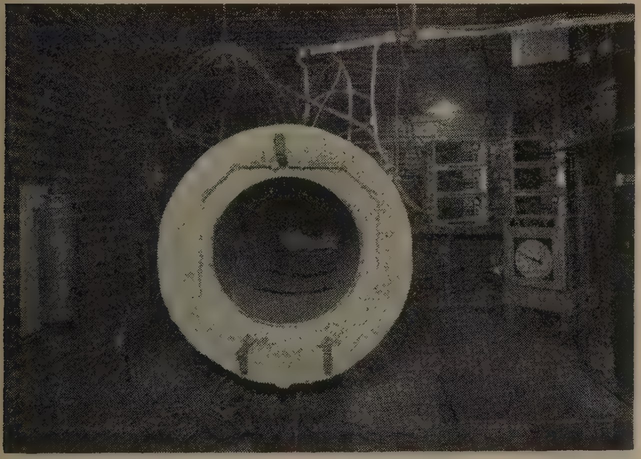

Fig. 6 Heating Furnace Showing Control Equipment

inserted at the same time, end to end, Fig. 6. A total of 24 thermocouples were installed and temperatures were recorded. "Normal" cooling was accomplished by the use of a blower connected to a perforated pipe inserted inside the test pieces, which may be seen in Fig. 7.

For the quench tests a steam-atomizing oil burner was used to introduce a fine water spray in the cooling-air stream, Fig. 8. Table 2 shows the heat cycles to which the test pieces were subjected. Fig. 9 shows a typical normal heating and cooling cycle. Fig. 10 shows a similar cycle during the quench tests. Figs. 11 and 12 show the temperature differences across the pipe wall during the quench tests. It will be noted that the temperature differential across the 1.625-in-thick pipe exceeded 400 F during the quench tests, whereas during the normal cooling tests the maximum differential did not exceed 50 F. Temperatures during the quench tests were difficult to control,

Fig. 7 End View of Heating Furnace Showing Specimen and Cooling Sleeve

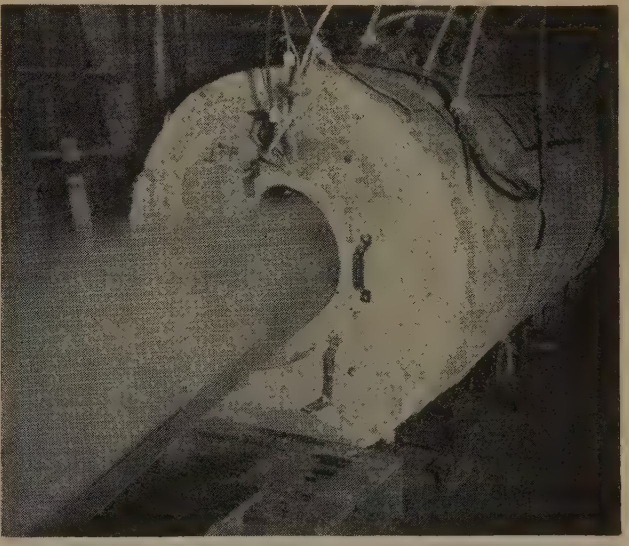

Fig. 8 Heating Furnace During Rapid Quench Test

TABLE 2 HEATING AND COOLING CYCLE PROCEDURE

	Cycles		
Heating and cooling	0–25	26–100	101–110
Heating period to 1100 F, hr	4	2	2
Holding period at 1100 F, hr	1½	0	0
Cooling period to 200 F, hr	2	2	...
Cooling period to 600 F	..	..	rapid

and therefore varied considerably. The most severe quench (upper range) and the least severe quench (lower range) temperature differentials across the pipe wall are plotted in the figures.

Results of Tests

Coupons were removed for examination before, during, and after the tests, and the entire test pieces were subjected to zyglo and x-ray examinations in accordance with the schedule shown in Table 3.

Zyglo examination of the multiweld piece after welding and before the heating tests indicated surface cracking in the 15/35

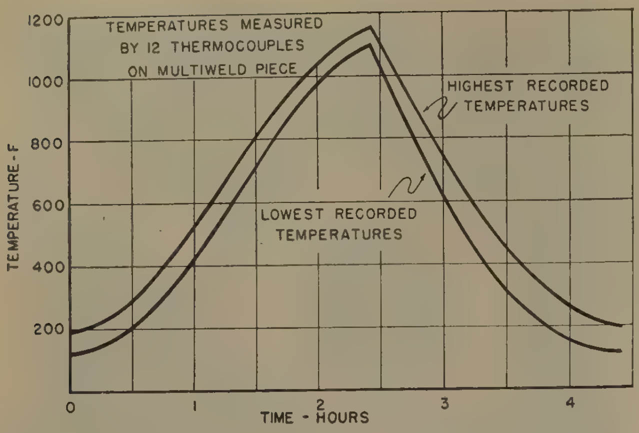

Fig. 9 Typical Heating and Cooling Cycle, Cooled With Forced Air, Cycles Numbers 26 to 100

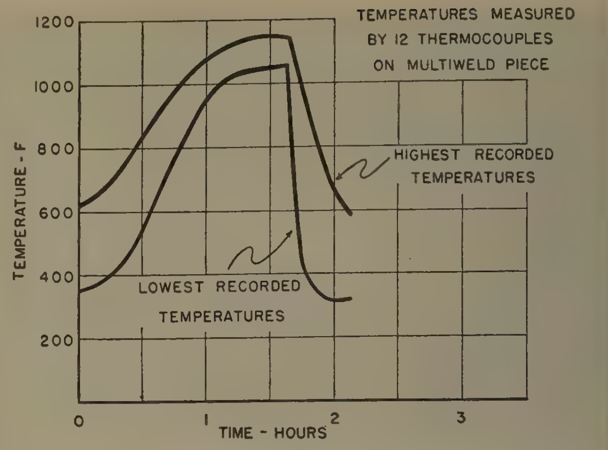

Fig. 10 Typical Heating and Cooling Cycle, Quench Cooling Cycles Numbers 101 to 110

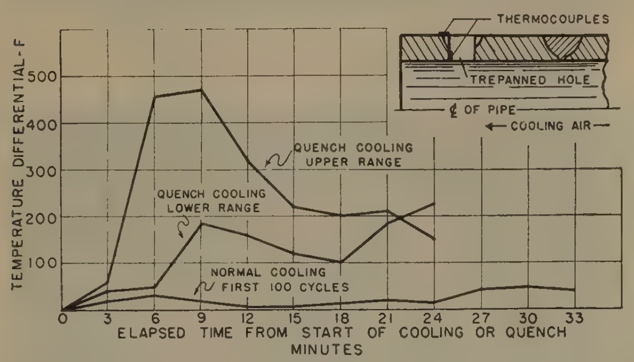

Fig. 11 Typical Temperature Differentials Outside to Inside of Pipe-Wall Multiweld Test Piece During Tests

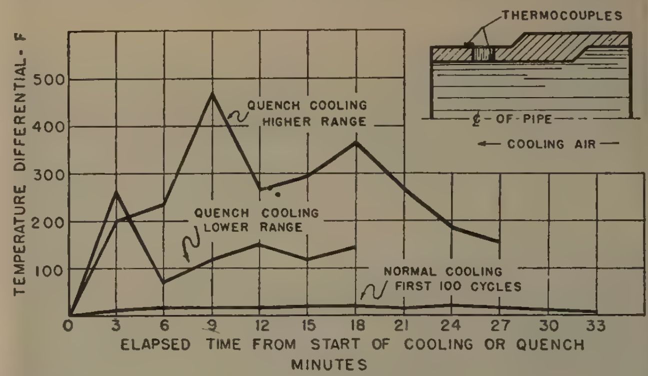

Fig. 12 Typical Temperature Differentials Outside to Inside of Kelcaloy Test Piece During Tests

TABLE 3 TEST EXAMINATION PROCEDURE

After cycle number	0	5	25	100	110
X ray	Yes	Yes	No	Yes	No
Zyglo	Yes	Yes	Yes	Yes	Yes
Trepan specimen	Yes	No	No	Yes	No
Microexamination	Yes	No	No	Yes	Yes
Physical tests	Yes	No	No	No	Yes

weld metal near the fusion line with the 18/8 base material over about 23 in. of the circumference, Fig. 13. This filler rod, not commonly used, was chosen because it had expansion characteristics about midway between the 18/8 and the $2^1/_4$ chrome materials. Cracking was also found in the 19/9 weld metal near the 25/20 buttered edge on the $2^1/_4$-chrome base metal. This extended over 9 in. of the circumference, as indicated in Fig. 14. The other three welds in the multiweld piece were sound, as determined by zyglo and x ray, as well as microexamination of the trepanned plugs, except for small weld-root cracks at the backing rings, which were present in all of the welds in the multiweld piece. Fig. 15 is a macrograph of the 19/9 Cb weld which is typical. Figs. 16 and 17 show typical microstructures. No defects were found in the Kelcaloy piece as fabricated. There were a number of ferritic inclusions in the 18/8 deposited metal due to improper control of the casting process.

Examination of the pieces during the tests and subsequent to the final quench tests, indicated that no new cracks developed as a result of the cyclic heating and cooling, and there was no indication that any of the cracks noted prior to the test due to unsound welds had increased to any measureable extent. Zyglo examination of the end of the Kelcaloy piece after the 100 cycles indicated an apparent separation between the two materials

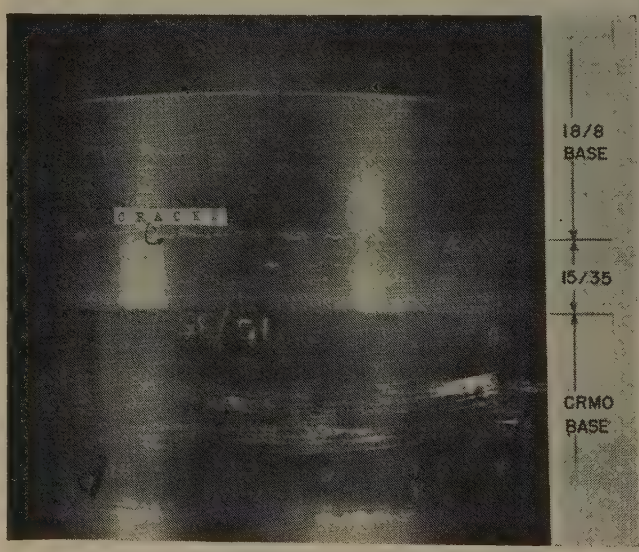

Fig. 13 Zyglo Examination of 15/35 Weld

at the surface at one end of the test piece. This could not, however, be verified by x-ray examination.

At the conclusion of the cyclic-heating tests, the test pieces were sectioned for detailed examination. Table 4 gives the results of the cold physical tests of the multiweld piece. Except for the 15/35 weld, which failed in the weld, all breaks in the

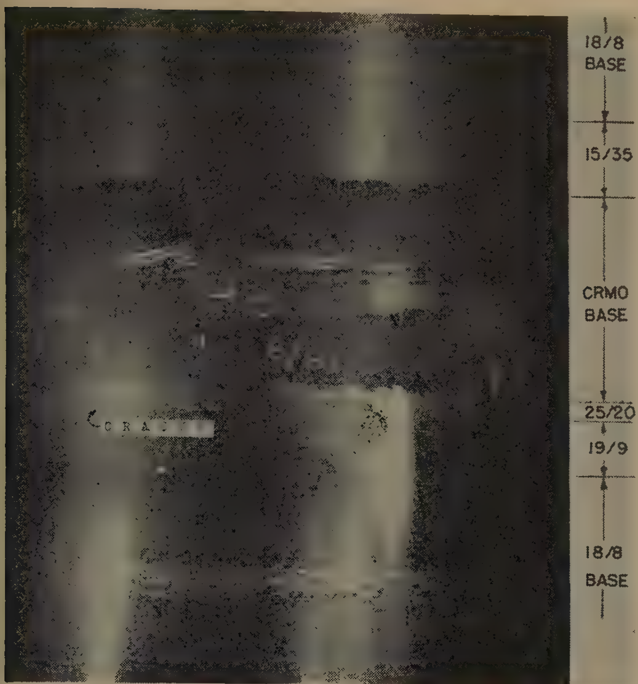

FIG. 14 ZYGLO EXAMINATION OF 19/9 WELD

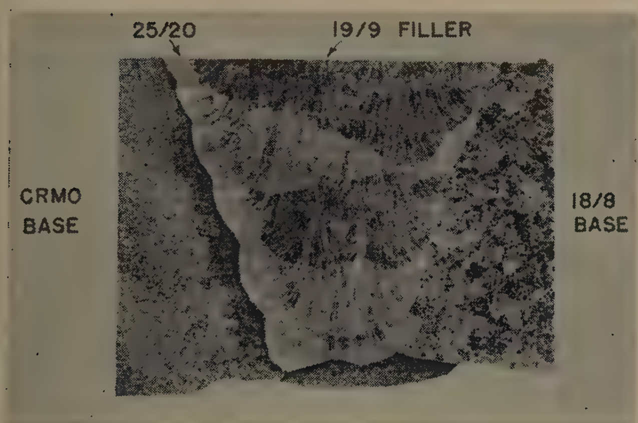

FIG. 15 MACROGRAPH OF 19/9 FILLER-ROD WELD

tensile specimens were in the $2^1/_4$-chrome base material. The cracking which had been indicated by the zyglo examination on two of the welds was found to be of negligible depth and was not detected in microexamination of cross sections of the welds. The bend-bar ductility of the specimens judged by ferritic material standards is low. All breaks in the bend bars occurred in the 18/8 base material or austenitic weld metal. It is expected that this matter, along with high-temperature rupture-test results will be the subject of further reports by others. It is difficult to make austenitic welds in heavy-wall pipe free from cracks. The welds tested were the first attempted, and the technique of welding had not been developed very far at the time. Thorough microexamination of the test pieces by several independent laboratories of the companies involved showed no discernible effect on the welds or parent metal which could be attributed to the cyclic heating and cooling.

Table 5 gives results of cold physical tests of the Kelcaloy piece. The junction between the austenitic and the ferritic

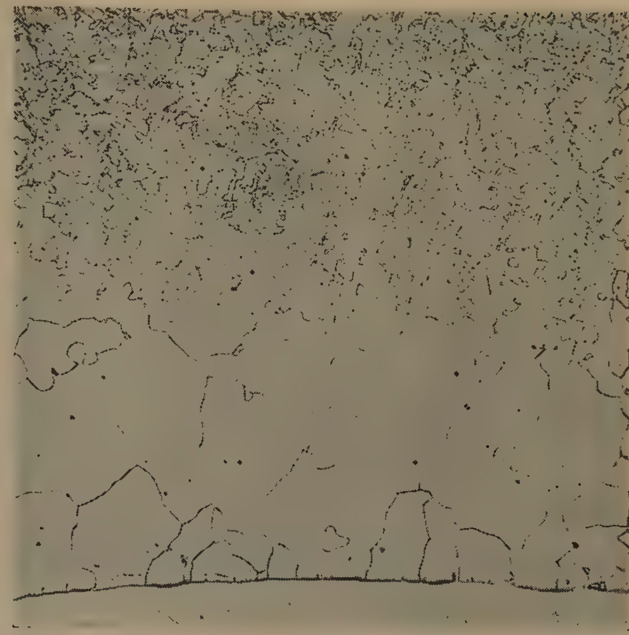

FIG. 16 MICROSTRUCTURE OF $2^1/_4$ CR-1 MO FORGING IN HEAT-AFFECTED ZONE ADJACENT TO 25/20, CB OVERLAY WELD; ×100

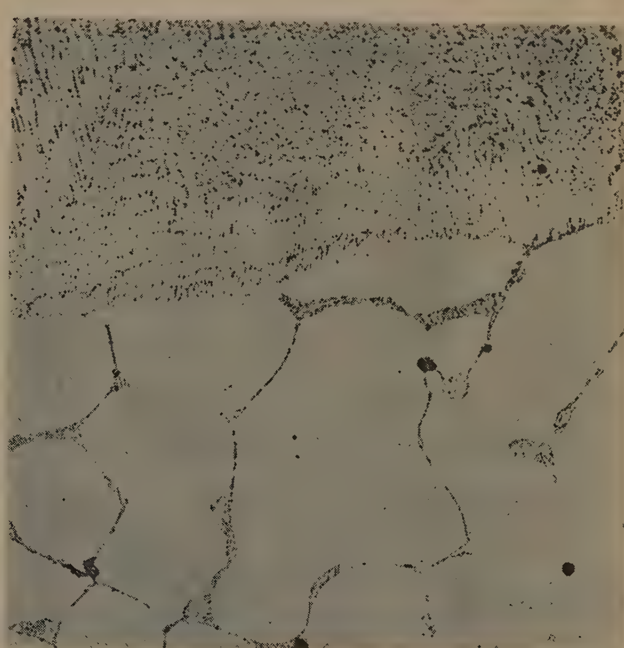

FIG. 17 MICROSTRUCTURE OF 18/8, CB CASTING AT FUSION ZONE WITH 19/9, CB WELD METAL; ×100

material was found to be entirely sound as judged by these tests. Bend tests of coupons removed from the Kelcaloy piece at the joint, where an indication of separation had been noted by zyglo examination previously mentioned, showed good ductility, and photomicrographs indicated that the apparent separation was due to intergranular oxidation of negligible depth. The ferritic inclusions which were found in the Kelcaloy piece proved to be weak points, as indicated by failure in those areas in bend-bar tests. The occurrence of these ferritic inclusions has been experienced before, and studies have shown that they can be

TABLE 4 RESULTS OF PHYSICAL TESTS ON MULTIWELD PIECE AFTER CYCLIC HEATING AND COOLING TESTS

(A) Transverse Tensile Tests Across Welds: (Reduced section—0.505 in diam)

Filler material	Per cent reduction in area, Top of weld	Per cent reduction in area, Bottom of weld	Per cent elongation in 2 in., Top of weld	Per cent elongation in 2 in., Bottom of weld	Ultimate, psi, Top of weld	Ultimate, psi, Bottom of weld
15/35	8.9	17.5	5.5	12.5	61900	65100
19/9	73.5	75.1	4.0	12.0	68200	68800
18/13	74.3	74.9	16.0	24.0	67200	67000
23/13	74.3	75.2	19.0	21.5	66200	67100
25/20	75.5	75.9	17.5	23.0	66200	66400

NOTE: All failures in $2^1/_4$ chrome base material except 15/35 which failed in weld.

(B) Free Bend Tests:

Filler material	Per cent elongation, Face	Per cent elongation, Root	Remarks, Face	Remarks, Root
15/35	..	..	Broke in preliminary bend in weld	Broke in preliminary bend in weld
19/9	..	..	Broke in preliminary bend in weld	Broke in preliminary bend in weld
18/13	25	..	Broke outside weld in 18/8	Broke in flaw in preliminary bend
23/13	..	..	Broke in weld at beginning of bend	Broke in preliminary bend in 18/8
25/20	8.7	1.5	Broke at edge of weld at start of face bend	Broke in 18/8 base material before bend completed

NOTE: All failures in 18/8 base material or welds.

(C) Side Bend Tests:

Filler material	Remarks
15/35	Cracked at beginning of bend in weld
19/9	Broke in weld at start of bend
18/13	Broke in weld at start of bend
23/13	Broke at edge of weld at start of bend
25/20	Broke at edge of weld at start of bend

NOTE: All failures in welds.

eliminated by close regulation of the electrical condition during the intermelting operation. It is expected that additional information on the physical tests of this piece also will be presented by others.

For the first three units at Sewaren it was decided to use Kelcaloy transition pieces between the ferritic piping and austenitic turbine stop valves. A cross section of the final design is shown in Fig. 18. In view of the satisfactory performance of the austenitic welds in the multiweld piece, a regulation butt weld using 19/9 Cb filler metal may be used for the fourth unit.

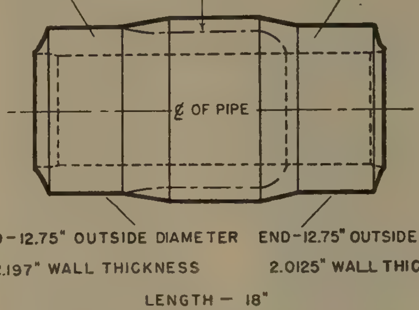

FIG. 18 ARRANGEMENT OF KELCALOY PIECE AS FINALLY DESIGNED

It is planned to continue the cyclic testing of full-size pipe of various materials and types of welds alongside the actual turbine installation at full pressure and temperature.

CONCLUSION

One hundred cycles of alternate heating and cooling of several

TABLE 5 RESULTS OF PHYSICAL TESTS ON KELCALOY PIECE AFTER CYCLIC HEATING AND COOLING TESTS

(A) Tests Between 18/8 Outer Layer and $2^1/_4$ Cr-1 Mo Inner Layer:

(a) TRANSVERSE TENSILE TESTS ACROSS JUNCTION

Specimen	Diameter, in.	Yield point, psi	Tensile strength, psi	Per cent elongation in 1 in.	Per cent reduction in area	Remarks
1	0.375	44600	67300	30	70	Ragged cup fracture in Cr-Mo
2	0.376	44200	66200	17.9	18.9	Straight fracture in 18/8

(b) SHEAR TESTS THROUGH JUNCTION

Specimen	Shear strength, psi	Remarks
1	59600	Sheared $^1/_2$ in 18/8 and $^1/_2$ in Cr-Mo
2	46000	Sheared in Cr-Mo

(c) SIDE BEND TESTS ACROSS JUNCTION
($^1/_8$ in. × $^1/_2$ in. bars)

Specimen	Per cent elongation
1	34
2	28

(B) Tests of Control Weld $2^1/_4$ Cr-1 Mo to $2^1/_4$ Cr-1 Mo:

(a) TRANSVERSE TENSILE TESTS ACROSS WELD
(Reduced section—0.505 in. diam)

Specimen	Yield point, psi; 0.5 per cent elongation, Top of weld	Yield point, Bottom of weld	Per cent Reduction in area, Top of weld	Per cent Reduction in area, Bottom of weld	Per cent elongation in 2 in., Top of weld	Per cent elongation in 2 in., Bottom of weld	Tensile Strength psi, Top of weld	Tensile Strength psi, Bottom of weld
1	...	37375	71.4	71.2	16.5	22.5	66000	64500
2	41750	38250	73.9	71.6	15.5	25.0	67000	64750

NOTES: All breaks in $2^1/_4$ chrome base material.

(b) FREE BEND TESTS $2^1/_4$ Cr-1 Mo CONTROL WELD

Specimen	Per cent elongation, Face	Per cent elongation, Root	Remarks, Face	Remarks, Root
1	44.2	31.4	Bent till ends met; two very small holes	Bent to 180 deg; one hole
2	41.5	36.0	Bent to 195 deg; one small hole in weld	Bent till ends met; one small hole in weld

(c) SIDE BEND TESTS $2^1/_4$ CR-1 MO CONTROL WELD

Both specimens passed satisfactorily.

full-size pipe joints between austenitic and ferritic steels through the range of temperature expected in normal operation, plus 10 cycles simulating water-carry-over conditions, produced no apparent deleterious results. The effect of long-time exposure at the operating pressure and temperature of course cannot be evaluated in a short-time test. It is believed, however, that the results provide considerable assurance that 18/8 piping, and particularly welds between 18/8 piping and low-chrome steel piping, can satisfactorily withstand the temperature changes which may be expected in power-plant service.

Acknowledgments

The multiweld piece was welded by the General Electric Company; chrome-moly sections were furnished by Public Service Electric and Gas Company; 18/8 castings and Kelcaloy piece were furnished by Westinghouse Electric Corporation. Advice and co-operation of the following individuals is acknowledged: E. L. Robinson, R. L. Jackson, and R. W. Clark of the General Electric Company; N. L. Mochel of the Westinghouse Electric Corporation; D. B. Rossheim, H. S. Blumberg, W. B. Bunn, W. Mc. McKee of the M. W. Kellogg Company; W. H. Armacost, F. I. Epply, K. Svendsen of Combustion Engineering Company; G. Dolan, United Engineers & Constructors Inc.; and F. P. Fairchild, H. M. Soldan, H. J. Robar of Public Service Electric and Gas Company.

Discussion

R. K. Akin.[2] Having been favored with reports of the progress of this investigation, we decided to conduct a parallel set of tests on tubing, the class of material with which we are more concerned. A report of our observations may be of interest as supplementary to the paper. First, however, will the author favor us with answers to the following questions:

1 After the multiweld piece was welded, was it allowed to cool from preheat temperature before stress-relieving, or was it raised immediately to the higher temperature?

2 Was the joint, made by the Kelcaloy process, stress-relieved in the same manner as the multiweld piece, and if not, what was the procedure?

Our investigation, like that of the author, was made to determine the effect on welded joints between ferritic and austenitic materials of alternate heating to 1100 F and cooling. We desired further to know whether rapid cooling from operating temperature would affect the joint differently from slow cooling. For the reasons offered by the author, we repeated the slow cooling cycle 100 times, but we increased the number of rapid cooling cycles from 10 to 25 in order to intensify the effects of this type of cooling.

Our tests were made on 2-in-OD × 0.200-in-wall tubing, involving flash-welded and fusion-arc-welded joints between

3 per cent chromium, 1 per cent molybdenum, ASME Spec 213, T-21

18 per cent chromium, 8 per cent nickel with titanium, ASME Spec SA 213, TP 321

25 per cent, chromium, 20 per cent nickel, 0.15 per cent maximum carbon

Arc welds were made using 3 Cr-1 Mo filler metal between 3 Cr-1 Mo and 18-8 Ti base metals, and 25–20 filler metal between the three combinations of base metals. Appropriate preheating and postheating was applied to each type of joint. Each type of joint was made in triplicate, one to be tested without cyclic heating and cooling, one to be tested after 25 cycles of heating to 1100 F, holding $1^1/_2$ hr, and water-quenching, and the third to be tested after 100 cycles of heating to 1100 F, holding $1^1/_2$ hr, and furnace-cooling to approximately 150 F.

The tests applied included the Zyglo test, Rockwell B hardness survey, free-bend tests, and microscopic examination. The Zyglo test was applied to determine first if there were any cracks in the original welds, and again after completion of cyclic heating and cooling to determine whether or not cracks had developed.

Hardness surveys were conducted to determine the efficiency of stress-relieving, the hardness characteristics before cyclic heating and cooling, and any changes which might have occurred because of cyclic heating and cooling.

Free bend tests were made to determine the ductility at various stages of thermal treatment and were considered the most important of the tests used. Tests were made on longitudinal sections following the spirit of the requirements of the boiler code, measuring the elongation of an original gage length equal to twice the tube-wall thickness.

All fusion welds involving 3 Cr-1 Mo material were preheated and maintained at 600 F minimum throughout the time of welding and until they were placed in a stress-relieving furnace. A preliminary stress relief at 1300 F for 1 hr was given to a group of welds before they were subjected to various experimental stress-reliving procedures. After a satisfactory procedure was adopted, newly welded samples were stress-relieved according to that procedure.

Hardness and bend tests were made on welds subsequently stress-relieved experimentally at subcritical temperatures and at various temperatures up to 1550 F, which was used by the author in his investigation. The data obtained are shown in Table 6 herewith. It will be noted that the hardness data are not very helpful in determining the properties of a weld. The ductility as shown by the free bend tests is more important. The welds with 1300 F preliminary stress relief failed by cracking with low elongation and can be considered as proof of the necessity for a proper stress relief.

A temperature of 1350 F, that is, subcritical temperature stress relief, produces better than 30 per cent elongation if the cooling rate is slow enough. Nonuniformity of furnace temperature, resulting in some overshooting, may cause some carbide solution so that air cooling from 1100 F tends to produce some hardening in the ferritic material.

Temperatures of 1500 F or higher cause excessive carbide migration with attendant weakening of the ferritic material through decarburization and embrittlement of the austenitic material through increase in carbide content and subsequent carbide precipitation. As a result, the bend specimens failed at or near the juncture of the two types of materials.

On the basis of these data we decided to stress-relieve, or "anneal" all joints involving this ferritic material by holding for 2 hr at 1350 F, cooling at a controlled rate, 50 deg F per hr, to 1200 F, and cooling further in the furnace to 800 F or less. Additional sets of joints 6 and 7 were further annealed by heating to 1550 F for 2 hr, cooling quickly in the furnace to 1250 F, holding for 1 hr, furnace-cooling to 1000 F, and then air-cooling.

Joints involving only austenitic materials were tested either as welded, or after annealing, or "equalizing," by holding at 1550 F for 5 hr and air-cooling.

The Zyglo test showed no cracks either before or after cyclic heating and cooling.

Table 7 shows the results of free bend tests, each set of figures representing four tests. The type of weld, materials joined, and the stress-relieving or annealing treatment are indicated. Samples tested prior to cyclic heating and cooling are designated by the letter "A." Those cooled rapidly from cyclic heating temperature are indicated by "R," and those cooled slowly, "normally,"

[2] Metallurgist, The Superheater Company, East Chicago, Ind.

TABLE 6 EXPERIMENTAL STRESS-RELIEF TREATMENTS: ROCKWELL B HARDNESS AND FREE BEND TESTS

Specimen no.	4A-0	4A-8	4A-1	4A-14	4A-15	4A-2	4A-3	4A-5	4A-6	4A-7	9A-10
Temperature, deg F	1300	1330	1350	1350	1350	1400	1450	1500	1550	1550	1350
No. of hours	1	5	2	2	2	2	2	2	2	2	2
Cooling rate, deg per hr	50	50	50	50	400	50	50	50	50	50	50
Minimum temp, deg F	1100	1200	1200	1200	800	1200	1200	1200	1200	1100	1100
No. of hours	0	0	0	0	2	0	0	0	0	0	0
Cooled in	Air	Air	Furn.	Furn.	Furn.	Air	Air	Air	Air	Furn.	Air
Hardness Tests:											
3 Cr-1 Mo tube	66	70	66	69	70	56	63	70	68	67	71
	86	86	87	89	89	87	75	71	71	69	87
	87	79	86	84	85	92	69	66	x	65	85
3 Cr-1 Mo filler	99	95	91	98	99	82	94	111	109	112	101
	110	97	91	96	101	101	101	112	110	112	103
	108	98	89	95	100	90	98	111	103	112	102
18-8 Ti tube	..	73	..	70	76	78	71	70	74	75	71
	77	75	68	71	72	73	75	72	74	79	77
	76	77	73	74	74	71	73	72	74	73	76
Bend Tests:											
NF or F	F	F	NF	NF	NF	NF	NF	F	F	F	F
Elongation, per cent	5.2	53.3	51.2	34.9	31.8	31.5	32.6	2.4	6.0	2.5	24.0
Remarks	Broke at 18-8 Ti weld	180 deg flat gas hole	180 deg flat	Good	Good	180 deg flat	180 deg flat	Broke at scribe mark	3M shear at weld line	Crack in over-lay	3 M shear at weld line

TABLE 7 PER CENT ELONGATION ON FREE BEND TESTS

Type of weld	Flash welds											
Joint no.	1 Stress relieved			2 Stress relieved			3 As welded			3 Annealed		
Materials	3 Cr-1 Mo—18-8 Ti			3 Cr-1 Mo—25-20			18-8 Ti—25-20					
Treatment	A	R	N	A	R	N	A	R	N	A	R	N
Maximum	40	26	30	44	52	40	42	42	40	44	46	46
Minimum	18	14	20	40	44	20	22	16	14	32	37	22
Average	31	20	26	42	47	34	28	26	26	37	42	38

Type of weld	Arc welds								
Joint no.	4 Stress relieved			5 As welded			5 Annealed		
Materials	3 Cr-1 Mo—18-8 Ti[a]			18-8 Ti—25-20[b]					
Treatment	A	R	N	A	R	N	A	R	N
Maximum	42	40	22	49	46	38	39	47	43
Minimum	30	30	12	41	43	34	33	33	40
Average	37	35	15	45	45	36	36	41	41

Type of weld	Arc welds											
Joint no.	6 Stress relieved			6 Annealed			7 Stress relieved			7 Annealed		
Materials	3 Cr-1 Mo—18-8 Ti[b]						3 Cr-1 Mo—25-20[b]					
Treatment	A	R	N	A	R	N	A	R	N	A	R	N
Maximum	29	32	32	33	32	28	33	36	49	34	32	30
Minimum	27	23	21	27	25	24	31	32	39	27	22	22
Average	28	27	25	29	27	26	32	34	43	30	28	28

[a] 3 Cr 1 Mo filler metal.
[b] 25–20 filler metal.

by "N." Hardness tests are not included because there were no changes observable due to the heating and cooling cycles. Joints 6 and 7 showed a difference in hardness of ferritic material as a result of prior thermal treatment. The stress-relieved samples had a hardness from 83 to 89 next to the deposited metal, and the annealed samples had a hardness from 72 to 75 at the same location. The austenitic material showed no difference in hardness.

A study of the data in Table 7 reveals the following:

1 Flash welds between 3 Cr-1 Mo and 18-8 Ti, stress-relieved at 1350 F, show comparatively low ductility especially after rapid cooling cycles.

2 Flash welds between 3 Cr-1 Mo and 25-20, stress-relieved at 1350 F, have good ductility, but show some loss after 100 cycles of slow cooling.

3 Austenitic filler metal of 25-20 composition produces a joint between 3 Cr-1 Mo and 18-8 Ti that retains more ductility, especially after the slow cooling cycles, than 3 Cr-1 Mo filler metal.

4 Joints between 25-20 chrome-nickel alloy and 3 chrome 1 moly are superior to those between 18-8 Ti and this ferritic alloy.

5 A stress-relieving temperature of 1350 F applied to a 3 Cr-1 Mo-25-20 joint is superior to a 1550 F annealing treatment, because of the greater ductility retained after cyclic heating and cooling.

6 Cyclic heating and cooling does not appear to detract from ductility of stress-relieved joints between 3 chrome 1 moly and 25-20 chrome-nickel.

7 With respect to ductility of a joint between the two kinds of austenitic materials, neither the "as-welded" joint nor the annealed joint shows superiority over the other, and joints in either condition retain good ductility after cyclic heating and cooling.

8 The rate of cooling does not appear to influence the ductility of the joint so much as the number of cycles. This may be an indication of the effect of time at operating temperature which these short-time tests cannot evaluate.

Based on the results of these tests, it would appear that joints between 3 per cent chrome 1 per cent molybdenum and 18-8 Ti materials would be improved if an intermediate section of 25-20 material were inserted. This type of construction has a threefold advantage: (1) The joint between ferritic material and 25-20 using 25-20 filler metal can be stress-relieved at 1350 F, holding carbide migration to a minimum, and then cooled appropriately to produce a ductile joint which remains ductile after several

cycles of heating to 1100 F and cooling. (2) The joint between 25-20 and 18-8 Ti, using 25-20 electrode, is easily stress-relieved by heating and holding at about 1550 F and then air-cooling, or it can be left as welded, either of which is satisfactory. The first type, requiring more precise stress-relieving temperature control, can be made in the shop; the second type between two austenitic materials does not require too precise control of preheat or post-heat so can be handled easily in the field. (3) The insertion of 25-20, having an intermediate coefficient of expansion between the other two alloys, supplies a structure less susceptible to the effects of differential expansion.

H. S. Blumberg[3] and W. B. Bunn.[4] This paper is timely in view of the relatively recent and growing interest in the use of the austenitic stainless steels as steam power-plant piping materials. The subject is a challenging one, and focuses on the need for considerably more knowledge regarding the behavior of ferritic and high-chromium high-nickel austenitic steels in elevated-temperature service. Of value in the reported study is the fact that the specimens tested actually represent the dimensions and design of piping which is to be used in service.

The author mentions that, in addition to the multiweld butt-welded structure which he describes in detail, a specimen of Kelcaloy was fabricated so that a comparison could be made between the two types of construction. A description of the manufacture of the Kelcaloy transition piece was not included in the paper, and therefore it will be given here.

Kelcaloy is the commercial name for the material made by the Kelcaloy process, which provides an automatic means for melting and solidifying metals progressively, so that relatively large objects can be produced. Essentially, the process is carried out in the Kelcaloy machine in which commercial raw materials in purified form are melted in a novel manner in special molds at a controlled rate through an electric arc under a highly protective flux. The liquid metal is progressively and rapidly solidified to produce metals of high uniformity and particular characteristics. Several thousand ingots of a wide variety of compositions have been made by this process during the past decade.

The Kelcaloy machine operates entirely automatically; it is at present approximately 40 ft in height. Raw materials enter at the top and move downward. These raw materials consist of ferro or other alloys and pure metals, flat metal strip, and slag. The alloying elements are commonly available in controlled-particle size; the strip is selected from commercial sources, and the slag is of special composition, In operation, the entering flat strip in coil form is moved downward at a controlled rate through rolls to form a tubular cross section; simultaneously, alloys are metered accurately through the moving tube by unique mechanical means; the tube continues to move uniformly downward and the alloys fall through it, so that an exact quantity of raw materials (strip and alloys) reaches the electric arc at the end of tube in a given time, thus making available at the arc within the mold a known quantity of liquid metal of controlled composition. The arc operates under a protecting blanket of molten flux. In the process there is continually present during melting a liquid level head. Thus there is neither pipe nor voids in the cast ingot, and a minimum of segregation. The ingot stress-relieves itself during the process so that even extremely air-hardening steels including high-speed steels can be made without cracking; subsequent heat-treatment is of course carried out if necessary. The range of analyses which can be made by this process depends on the availability of raw materials in proper form.

The process will make objects of uniform composition in a single melt, or it may be utilized to produce integral objects of dissimilar metals. Thus, in making clad-steel plates, slabs or ingots of base metal are set up so that one face of the latter forms one of the four sides of a rectangular mold. The molten material is manufactured in the mold space, and provision is made for intermelting simultaneously about 1 in. into the slab or ingot face. Because speeds of diffusion of metallic elements are extremely high at temperatures reached by the molten metal, the clad layer is uniform in composition across its entire thickness. An integrally clad object is thus available to the steel mill which can be rolled or otherwise hot-worked in conventional manner to produce clad plates, sheet and strip, or it may be used in the "as-clad" condition.

The Kelcaloy transition piece is a clad material. In its manufacture, an annealed $12^1/_2$-in-diam forging, 30 in. long, of low-carbon $2^1/_4$ per cent Cr-1 per cent Mo composition, furnished to ASTM Spec A-213, Grade T22, served as the base metal. This forging was centered in a round water-cooled copper mold, so that a space of approximately 2 in. was present between the outer surface of the forging and the inner face of the mold; 18-8 Cb austenitic steel was manufactured and intermelted with the face of the $2^1/_4$ per cent Cr-1 per cent Mo forging. The strip used as one of the raw materials for melting was of plain-carbon-steel composition, and alloys consisted of low-carbon ferrochromium, ferromanganese, ferrosilicon, ferrocolumbium, and nickel. The clad piece was annealed by heating to 1550 F, holding for 4 hr, followed by furnace-cooling. It was then machined to the dimensions shown on the Kelcaloy portion of Fig. 4 of the paper. It will be noted that a $2^1/_4$ per cent Cr-1 per cent Mo spool piece was welded to the $2^1/_4$ per cent Cr-1 per cent Mo end of the transition piece with electrodes of similar analysis.

The author reports the presence of "ferritic inclusions" in the Kelcaloy section when it was cut up for testing. These small crescent-shaped inclusions consisted of low-carbon steel, and were the result of electrical difficulties encountered during Kelcaloy manufacture of the transition piece. Electric meters in the Kelcaloy machine are used to give a clear indication as to arc stability and uniform melting. In the Kelcaloy manufacture of the transition piece, it was known that these inclusions were being formed during a short part of the melting cycle. Ordinarily, such a Kelcaloy piece is not accepted by our inspection department. However, when an attempt was made to obtain another $2^1/_4$ per cent Cr-1 per cent Mo core section, it was learned that there would be considerable delay. In a review of the situation, it was decided that these inclusions, because of their small size and number, would probably not affect the behavior of the Kelcaloy section in the cycling tests.

Subsequent to the manufacture of the transition piece used in the presently discussed tests, four more were made for actual use at Sewaren. No difficulties were encountered in Kelcaloy intermelting. Careful observations were made during machining of the alloy surfaces, and this was compared with the results of x ray, Zyglo, and magnaflux testing. Nod effects of any kind were observed in any of the four sections.

It has also been requested that the results given in Table 5 of the paper which were made in our laboratory, be discussed in somewhat more detail.

(*A*) lists data obtained on sections removed from the Kelcaloy piece, after cyclic heating; the physical test results represent values for the junction and vicinity of the $2^1/_4$ per cent-Cr Mo and 18-8 Cb integral transition piece.

(*A*) (*a*) This gives values obtained after testing 0.375-in-

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diam tension specimens with the Kelcaloy junction at the center of the test bars; one half of the length of the specimen consisted wholly of the ferritic $2^1/_4$ per cent Cr-Mo material, the other half of the bar of manufactured austenitic 18-8 Cb steel. A 1-in-gage length, $^1/_2$ in. on each side of the junction, was marked on each test specimen. Two tests were made, one fracturing in the $2^1/_4$ per cent Cr-Mo portion near the junction, with good elongation and reduction of area. The other specimen broke in the austenitic portion, with lower elongation and reduction of area. There was little difference in tensile strength and yield points between the two bars.

(*A*) (*b*) This shows the results of shear tests. These specimens were prepared in accordance with Fig. 1, ASTM A264-44T, in which a small rectangular button of manufactured austenitic steel of fixed dimensions is present above the base metal, so that shearing force can be applied in testing by utilizing a jig and a conventional tensile testing machine. Since load and area in shear are known, calculation can be made in terms of shear strength. Values of 59,600 and 46,000 psi were obtained in two tests. These values are approximately equal to those for unitary materials, that is, if specimens were tested wholly of either $2^1/_4$ per cent Cr-Mo or 18-8 Cb material. It is interesting to note that the shear-test requirement of the ASTM specification for clad steels is a minimum of 20,000 psi.

(*A*) (*c*) This gives values of side bend tests in which the cross section of the wall of the Kelcaloy transition piece was made the outer fiber of the side bend test. Again, one half of the bar consisted wholly of $2^1/_4$ per cent Cr-1 per cent Mo steel, the other half of 18-8 Cb. The junction of the two dissimilar metals was made the center of the outer fiber of each specimen. No failures were observed when the two specimens were bent to 34 per cent and 28 per cent respectively, at 180 deg.

Test values reported in (*B*) refer to specimens cut from the welded ferritic end of the transition piece after cyclic testing. Each of the tested bars contained a $2^1/_4$ per cent Cr-Mo weld in the center which joined the $2^1/_4$ per cent Cr-Mo transition piece end to a length of $2^1/_4$ per cent Cr-Mo tubing as described in the paper. Transverse tension bars containing the weld in the center and base metals at each side gave good strength and ductility. The elongation values reported are not particularly significant, in view of localized nonuniform yielding due to the presence of weld, heat-affected zone, and unaffected base metals, all within a single gage length. Free-bend and side-bend test values reported in (*B*) (*b*) and (*B*) (*c*) were entirely satisfactory. All tests values reported in (*A*) and (*B*) are indicative of excellent bond quality in the transition piece and of $2^1/_4$ per cent Cr-Mo weld characteristics.

Table 4 of the paper reports results of physical tests on the five welds of the multiweld piece after cyclic testing. Ultimate strength values are satisfactory, but a relatively low order of ductility is indicated in these welds, as judged by face, root, and side-bend tests. The transverse-tension elongation values reported in (*A*) are difficult to interpret, when it is considered that deposited weld metal, two heat-affected zones, and two unaffected base-metal regions in dissimilar metals were all contained within the single 2-in-gage length of each 0.505 bar. Obviously, there was considerable opportunity for highly localized deformation so that the reported elongation figures cannot be evaluated reasonably. It is to be noted that except for the 15 Cr-35 Ni specimens which failed in the weld, all test pieces broke in the $2^1/_4$ per cent Cr-1 per cent Mo base-metal portions. The high reduction of area is indicative of the excellent ductility of the $2^1/_4$ per cent Cr-Mo material.

A résumé of the reported face, root, and free-bend failures is given in Table 8 herewith. The outstanding facts in Table 8 are the preponderance of failures in the weld and the extremely lo ductilities obtained.

TABLE 8 SUMMARY OF FACE, ROOT, AND FREE BEND TEST REPORTED BY THE AUTHOR FOR MULTIWELD PIECE WELD AFTER CYCLIC TESTING

	Face bend test		Root bend test		Side bend test	
Joint	Elongation, per cent	Broke in	Elongation, per cent	Broke in	Elongation, per cent	Broke
15-35......	0	Weld	0	Weld	0	Weld
19-9 Cb....	0	Weld	0	Weld	0	Weld
18-13-2 Cb.	25	Type 347	0	Weld	0	Weld
23-13 Cb...	0	Weld	0	Weld	0	Weld ed
25-20 Cb...	9	Weld edge	2	Type 347	0	Weld ed

In order to obtain some indication of the relative ductilities of the various zones in each of the five welds, our laboratory, afte discussions with the author and his associates, prepared a numbe of miniature bend-test specimens from each weld and centere each in testing so that maximum bending would be located i the weld, the heat-affected zones in each base metal, and in th unaffected base metals of each of the five welds. Because o the limited material available, small specimens had to be usec Results are given in Table 9 of this discussion. The salien points developed by these results are as follows:

(*a*) Relatively low ductility of all welds.
(*b*) Relatively low ductility of all austenitic junctions, wit sharp failures at the junction lines.
(*c*) Good ductility of all ferritic junctions.
(*d*) Low ductility of austenitic base metal.
(*e*) High ductility of ferritic base metal.

From this it appears that low-ductility regions are confined to each of the five welds and the austenitic junctions.

An attempt was made to correlate these results in terms of:

1 Base-metal chemistries.
2 Weld-metal chemistries.
3 Welding history.
4 Heat-treatment.
5 Cyclic test conditions.
6 Microstructures of base metals and welds.
7 Results reported by previous investigators in this field.
8 Heavy-wall welds made and tested in our laboratories.

The 18-8 Cb and $2^1/_4$ per cent Cr-Mo base metals have satisfactory chemical compositions; these are consistent with the requirements of ASTM specifications. Microstructure of the austenitic material was coarse grained, typical of cast material, with eutecticlike aggregate at grain boundaries. The $2^1/_4$ per cent Cr-Mo base metal was well spheroidized.

Weld-metal chemistries appear to be typical of those to be expected in each particular weld. In view of recent researches reported, it appears that carbon contents of all welds are in that range, reported by Thomas and co-workers, which give low ductilities. However, the actual elongation values reported in the full-sized specimens are so low that they do not appear to be related to chemical analyses.

No factor was indicated in the welding history which might account for the bend-test results. Some welding engineers feel that low preheat or perhaps complete absence of preheat with controlled low interpass temperature is desirable. There is, however, as yet no unanimity of opinion on this point.

Microscopic studies were made of full cross sections of each weld. In general, the welds were reasonably free of nonmetallics; however, the 23-13 Cb and 25-20 Cb welds were noted to contain some slag stringers of the type described by Carpenter. All welds were expectedly coarse grained, but no microcracks were observed. It was not possible in our studies to relate the low ductility values to microstructure.

TABLE 9 ELONGATION VALUES OBTAINED IN TESTING MINIATURE BEND SPECIMENS FORCED TO BEND AT LOCATIONS INDICATED BEFORE AND AFTER CYCLIC TESTING

	Before cyclic heating		After cyclic heating									
					Junction				Base Metal			
	Weld		Weld		Austenitic		Ferritic		Austenitic		Ferritic	
Joint	Elong, per cent	Broke in	Elong, per cent	Broke in	Elong, per cent	Broke in	Elong, per cent	Broke in	Elong, per cent	Broke in	Elong, per cent	Broke in
15-35	22	Weld	12	Weld	20	Junct.	30	No cracks[a]	4 8	..	36 38	..
19-9 Cb (Buttered)[b]	10	Weld	14	Weld	4	Junct.	32	No cracks	..	..	..	..
18-13-2 Cb (Buttered)[b]	6	Weld	2	Weld	10	Junct.	24	No cracks	..	..	..	..
23-13 Cb	6	Weld	2	Weld	10	Junct.	40	Junct. & weld	..	..	..	..
25-20 Cb	10	Weld	4	Weld	10	Junct.	36	Junction	..	..	..	..

[a] Except for slight root bend crack.
[b] Buttered with 25-20 Cb weld metal on $2^1/_4$ per cent Cr-1 per cent Mo base-metal groove edge.

In a consideration of heat-treatment as a possible factor, it was realized that there are data to show that "stabilization" treatments, consisting of holding at 1550 F and 1650 F and air or furnace-cooling, causes some lowering of ductility. However, the low values obtained are not typical of the effect of such heat-treatments.

The miniature bend-test results on weld metal representing no cyclic treatment are of the same general order as after cyclic exposure. This would tend to indicate that the heating and cooling procedures were not a primary factor.

Table 10 herewith lists a comparison of the $1^5/_8$-in-thick 19-9 Cb weld (between dissimilar metals) reported by the author, and five $^3/_4$-in-thick welds made in our welding laboratory joining Type 347 base metals. The latter joints were restrained during welding by heavy fillets at the plate edges. The considerably higher elongation values than obtained in the 19-9 Cb weld reported in the paper are apparent.

Our studies thus far have disclosed no clear cause for the low elongation values reported.

It is possible that the poor results obtained are traceable to the heavy thickness welded, with which there is relatively little experience to date. The need for further experimental studies in heavy-walled tubes is clearly indicated.

The encouraging feature of the data presented by the author lies in the fact that no failure was encountered as a result of the cyclic heating and cooling.

This paper will undoubtedly serve as a strong stimulation to many investigators to carry out further studies to fill in gaps in the present knowledge.

TABLE 10 COMPARISON OF CHEMICAL ANALYSES, CR-NI EQUIVALENTS, AND FREE-BEND-TEST ELONGATIONS OF 19-9 Cb WELD REPORTED BY THE AUTHOR, AND FIVE WELDS MADE AT THE M. W. KELLOGG COMPANY

	Element	19-9 Cb weld reported by author	Welds made and tested by— No. 1	No. 2	No. 3	No. 4	No. 5
Chemical analysis	C	0.062	0.081	0.074	0.080	0.060	0.077
	Mn	1.80	1.67	1.45	1.46	1.61	1.46
	Si	0.68	0.60	0.42	0.60	0.42	0.41
	Cr	18.64	19.5	19.5	19.7	19.1	21.0
	Ni	10.78	10.6	10.6	10.5	9.8	10.0
	Cb	0.84	0.83	0.86	0.91	0.89	0.70
	$\frac{\text{Cr Eq.}}{\text{Ni Eq.}}$	1.64	1.64	1.64	1.70	1.76	1.80
Weld thicknesses, in.		$1^5/_8$	$^3/_4$	$^3/_4$	$^3/_4$	$^3/_4$	$^3/_4$
Free bend tests, per cent	As welded	...	31.3	42	41.7	41.7	37.8
Elong. in weld, per cent	1600 F 4 hr	0[a]	28.6	29.2	36.5 36.5	37.2	38.5

[a] 1550 F, 4 hr.
NOTE: (1) Weld reported by author joined dissimilar metals. M.W.K. Co. welds joined wrought Type 347 materials.
(2) All $^3/_4$-in-thick welds made under restraint.

P. M. BRISTER.[5] The tests reported give a good answer to one of the questions in making this type of weld, namely, that of differential expansion under cyclic temperature conditions. The tests have aided many of us in our studies of this problem since knowledge of the results has prevented duplication of test work and has permitted greater study of some of the other problems connected with welding ferritic materials to austenitic materials.

We have been doing a considerable amount of test work on this subject in our Works Control Laboratory in Barberton under Mr. Carpenter, and much of the information we have obtained is difficult to evaluate. It is apparent from our studies and tests that time is a very important factor, as well as stress and temperature, and we are sure much will be learned from the test specimen to be installed at Sewaren.

Our discussion of this paper is based on our development work and experience.

One of our tests is a rotating-beam fatigue test, with the test specimen being a heavy wall, $2^1/_2$-in-OD superheater tube. In testing welds between austenitic and ferritic tubes, we have found considerable difference between the results when tested with no internal pressure on the tube specimen, and with 2000 psi internal pressure. We at first thought that the pressure stress would be relatively small in comparison with the stresses from welding and differential expansion and would not affect the results appreciably. However, test specimens having internal pressure showed a considerably shorter fatigue life and indicated that the stress imposed by the internal pressure had an effect out of proportion to the value of the stress. Apparently, at a temperature level of 1050 F to 1100 F, the creep effect on the behavior of the materials is not apparent in evaluation of the stresses with the elastic theory. We believe there is much more to be learned on the behavior of metals under load in the creep range, and that laboratory work simulating service conditions should consider all loads from whatever source.

Our experience indicates that welding austenitic materials with preheat temperature as high as 600 F is not good practice. Some of the results to be expected were presented in a paper by O. R. Carpenter.[6] This paper indicated that austenitic materials should be welded with a low preheat and with low interpass temperatures in order to avoid the inclusion of intergranular slag envelopes in the weld metal with a consequent loss of ductility.

When a postweld heat-treatment of 1550 F is given a weldment having ferritic base metal and an austenitic weld metal, such as reported in this paper, considerable carbon migration from the ferritic steel to the austenitic weld metal results. This is apparent in Fig. 16 of the paper. There is also carbon enrichment at the line of fusion resulting from the migration. Evaluation of the effects of the weak zone of carbon depletion and possible embrittlement at the line of fusion under the operating conditions of such a weldment are very difficult. Our tests with the fatigue machine showed that the decarburized zone is the weakest part

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[6] "Some Factors Controlling the Ductility of 25 per cent Cr-2040 Ni Weld Deposits," by O. R. Carpenter, American Welding Society, October, 1947.

of the joint and that failures started at slight notches. However, we have examined a weld made with an austentitic rod between two pieces of cast 5 per cent chromium pipe which had 11,000 hr service at 1000 F, and 375 psi, and had the temperature cycled to room temperature seventeen times during this period. This 5 per cent chromium pipe had received a 2-hr 1575 F postweld heat-treatment, followed by air-cooling and drawing at 1250 F, and there was found to be appreciable carbon depletion in it. There were also numerous slag pockets and voids which could have acted as internal stress raisers. However, the weld did not fail in service, though bend tests on the weld indicated the defective condition of the weld.

While we do not believe the carbon migration in this type of weld with the type of service expected will result in an immediate failure or dangerous condition, we believe that, since it is the weakest part of the weld, it should receive considerably more study and should be watched carefully in service. It is possible that the effect of carbon migration in a joint, as shown in Fig. 18 of the paper, will be different from a more or less standard V-type weld groove.

We agree with the author that it is difficult to make welds in heavy-wall pipe free from cracks. It is necessary to employ careful techniques and to control the weld-metal analysis within proper limits. Lack of proper techniques and weld-metal analysis control can produce defects which are microscopic in nature and difficult to recognize. Our experience indicates that it is these defects which tend to produce notches that can eventually lead to failure.

In summary, we believe this paper has contributed much to the industry. We all recognize that the power industry is on the threshold of a new era of high steam temperatures, necessitating the use of materials not heretofore used extensively in this industry. We all have much to learn and it is through papers such as this that we can all benefit by interchange of information.

J. D. Conrad.[7] The tests described in this paper indeed give assurance to the designer that joints between austenitic and ferritic materials may be expected to stand up in service. However, such tests are necessarily limited in scope, in this case only one size and wall thickness of pipe being used. Therefore some analysis of the stresses in the test piece, and the manner in which these stresses vary with the proportions of the piece seems desirable.

The problem lends itself to at least an approximate analysis without undue complications. Let us consider a pipe joint which is located a sufficient distance from rigid parts or other joints as to be unaffected by them. Assuming the thickness of the pipe is relatively small compared to its diameter, a rather simple approach may be made by applying the well-known theory of the bending of a beam on an elastic foundation. The formulas derived by the application of this theory will be given. The following nomenclature applies to this discussion:

T_1 = temperature at which joint is fully stress-relieved
T_2 = temperature at which stress is computed
α_1 = coefficient of espansion of austenitic steel
α_2 = coefficient of expansion of ferritic steel
r = mean radius of pipe
h = thickness of pipe
Δ = difference in radial expansion or contraction of the two materials
x = distance along axis from junction to point of stress

$$\beta = \frac{1.285}{\sqrt{rh}}$$

$$D = \frac{Eh^3}{12(1-\mu^2)} \doteq 2.67 \times 10^6\, h^3$$

σ_b = bending stress
σ_t = tangential stress
τ = average shearing stress

$$\Delta = (T_1 - T_2)(\alpha_1 - \alpha_2)\, r$$

$$M_x = \Delta\beta^2 D e^{-\beta x} \sin \beta x$$

$$M_{\max} \text{ (for } \beta x = \pi/4) = 0.322\,\Delta\beta^2 D$$

$$\sigma_b = \frac{6\,M_{\max}}{h^2} = 8.5 \times 10^6\,(T_1 - T_2)(\alpha_1 - \alpha_2)$$

$$P_x = \Delta\beta^3 D e^{-\beta x}(\cos \beta x - \sin \beta x)$$

$$P_{\max} \text{ (for } \beta x = 0) = \Delta\beta^3 D$$

$$\tau = \frac{P_{\max}}{h} = 5.6 \times 10^6\,(T_1 - T_2)(\alpha_1 - \alpha_2)\sqrt{h/r}$$

$$\sigma_t = \frac{E\Delta}{2r} e^{-\beta x} \cos \beta x$$

$$\sigma_{t\,\max} \text{ (for } \beta x = 0) = \frac{E\Delta}{2r} = 14.5 \times 10^6\,(T_1 - T_2)(\alpha_1 - \alpha_2)$$

It will be noted that the proportions of the pipe do not affect the maximum bending and maximum tangential stress, i.e., these stresses are independent of both wall thickness and pipe radius. The shear stress at the joint varies with $\sqrt{h/r}$.

However, the wall thickness and pipe radius have an appreciable influence on the extent to which all the stresses are carried along the axis of the pipe.

Fig. 19 of this discussion shows the stress variations plotted as a function of the quantity βx. As β varies with the proportions of the pipe, then the extent to which these stresses are carried along the axis of the pipe depends upon the relative wall thickness and pipe radius. Computing β and using these curves, the designer can determine readily a safe distance to keep such junctions from flanges, valve bodies, unremovable weld rings, or anything which might serve to raise these stresses.

The use of these curves also indicates that the $5^1/_2$-in. length of the sections in the multiweld test piece is such that a cross-effect of one junction on the other takes place. For this test pipe β is 0.43 and for an x value of $5^1/_2$, βx is 0.75π. Thus it is seen from the curves that each junction is within the influence of the adjacent ones.

Using the same general theory as was used in the derivations of the foregoing formulas and curves, an analysis of the actual

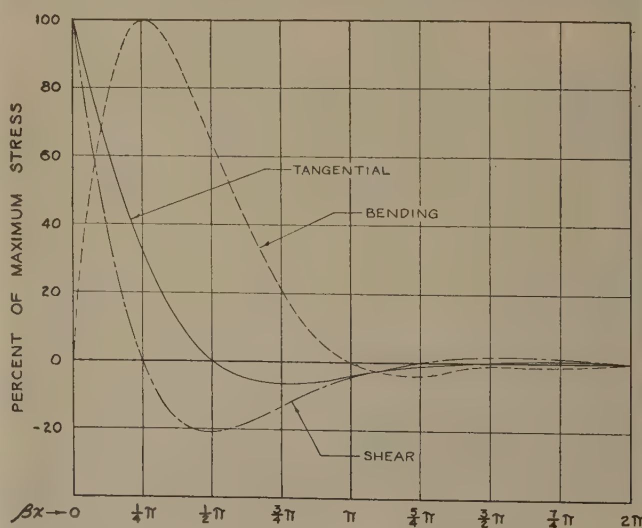

Fig. 19 Stress Variations Along Axis of Pipe

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stresses in the multiweld test piece was found feasible. The results are shown in Fig. 20 herewith.

These curves give the stress condition at room temperature and are calculated on the basis of no stress at 1550 F. It is assumed that the value of E at room temperature is 29×10^6 for both materials. The value of $(\alpha_1 - \alpha_2)$ is taken as 2.25×10^{-6}.

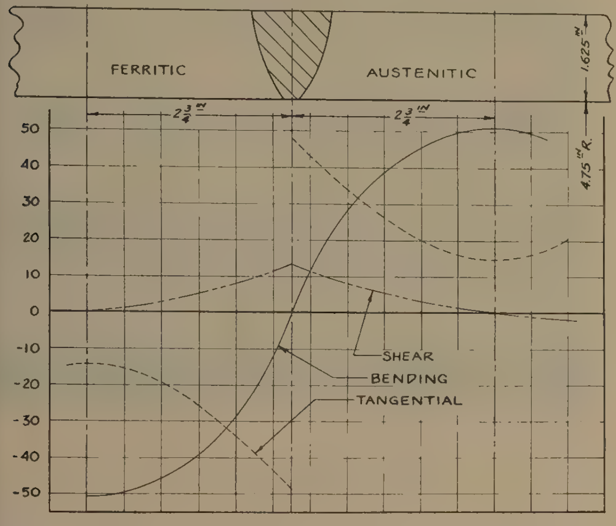

FIG. 20 STRESSES IN MULTIWELD TEST PIECE AT ROOM TEMPERATURE AFTER BEING FULLY STRESS-RELIEVED AT 1550 F

It is interesting to compare the maximum stresses in the test piece with the value computed for junctions sufficiently separated from each other as to avoid cross-effects. Bending stress is reduced from 51,000 to 29,000 psi, and shear stress from 13,000 to 10,000 psi. The tangential stress is unaffected.

The stresses so far discussed are, in effect, set up stresses at room temperature and, as such, are probably only of secondary importance. The change of stress through the cycling range is the chief criterion for failure. For the range of 200 F to 1100 F, we get a variation of 69 per cent of these stresses. The stresses at 1100 F are 20 per cent and at 200 F are 89 per cent, respectively, of the values on these curves. Therefore the cycling is between these two sets of stresses. The fact that the highest stresses are at room temperature is indeed a favorable condition.

There is a third stress component not so far considered. At the transition between the two materials there is a local radial compression on one side and an equal radial tension on the other. This stress has the same sign and numerical value as the tangential stress at the same location. In other words, we have equal biaxial tension and compression on either side of the material transition point. The third component of principal stress is zero at this point. Combined stress may be evaluated accordingly.

A stress analysis of the Kelcaloy joint has also been made. This work is more involved and its presentation is not within the scope of this discussion. In general, the theoretical stresses in this Kelcaloy piece are somewhat higher than in the junctions just described. This is particularly true when stress concentration, due to change in wall thickness within the piece, is considered. Other advantages as pointed out by the author, may, however, outweigh this disadvantage of higher theoretical stress.

ERNEST L. ROBINSON.[8] To the writer there seem to be two important lessons to be learned from the series of tests described by the author. The first lesson is the ease and success with which a welded joint may be made between the very different materials used for the station piping and the turbine piping. The other lesson is the need for getting a new background of judgment in the acceptance of materials selected for conditions never before employed.

A little more might be said with reference to each of these lessons. The success of making the welded joint between the unlike materials was not altogether unexpected, although nonetheless gratifying. Such joints had previously been used successfully in superheater tubing. They had also been used successfully in the turbine wheels of jet engines manufactured during the war. The ability to stand up under 100 such cycles as are anticipated in the Sewaren plant, including a smaller number of the severe cycles, has been demonstrated. How such joints will survive the conditions of service not immediately reflected in the tests is a matter in which we shall all be interested.

The materials chosen for use in this plant have been selected particularly because of their long-time strength at high temperature, as demonstrated in the most appropriate tests it has been possible to devise, among which are included those described by the author.

Over many years it has been customary to judge the materials used under less severe conditions by well-known tests indicating short-time strength and ductility under ordinary conditions.

The second lesson to be learned from this presentation is that tests, although they have been appropriate in years past for ordinary materials under ordinary conditions, can no longer be accepted as a suitable background for judgment of the new materials developed for the extraordinary conditions now being anticipated. It is not the writer's intention to underestimate the importance of any of the short-time low-temperature tests of strength and ductility. The important thing to note is that they were appropriate for use in judging the materials in current use when the test methods were developed. Wholly new levels or wholly new tests or both are required for the short-time acceptance of materials suitable for long-time service at extreme temperatures.

For instance, we know that large amounts of plastic strain may be withstood safely for a limited number of cycles, as this paper well shows. We are extending the well-known S-N fatigue curve far to the left and finding that it is possible to creep and uncreep successively many times under conditions which, on a stress basis, would look very severe. We know that the acceptability of these compositions is improved frequently by low-temperature hardness of a degree which would cast suspicion on the softer materials used at lower temperatures. The Sewaren installation and this paper constitute a step toward the determination of the limitations which must be known as we go to higher steam temperatures.

R. L. JACKSON.[9] From the results of the tests described in the paper, it was concluded that a welded joint between ferritic and austenitic steels showed no deleterious effects from the cyclic temperatures imposed upon it. This conclusion was based on the tests conducted over a relatively short time. To gain further knowledge of the properties of such welds, the writer's company is running long-time rupture tests on bars from each of the welds in the test piece, Fig. 1, in the paper. This discussion will cover those tests and will comment on the matter of ductility and other weld properties.

As stated in the paper, bend bar tests on samples cut from the welds showed little apparent ductility. This same lack of ap-

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[9] Turbine Engineer, Apparatus Department, General Electric Company, Schenectady, N. Y. Mem. ASME.

parent ductility has been encountered by the writer's company in welds between dissimilar materials before; one such case being a weld between $2^1/_4$ Cr-1 Mo steel and 1 Mo-0.2 V steel. In such cases, upset sections have been used to reduce the stress below that in straight pipe sections. When dissimilar materials are welded together and subjected to bending, most of the bending is concentrated in the more ductile of the two materials causing failure, with little apparent ductility indicated between the scribe marks. It may be that a new concept of what is obtainable in room-temperature properties will be formed when more information is available on the alloys being considered for service at 1000 F and up.

Early attempts at welding austenitic materials showed considerable cracking in the deposited metal. By continued development with attention to rod coating and chemistry, this trouble has been almost eliminated. Since the time of the heat-cycle tests, considerable welding has been done on the various parts of the Sewaren turbine built by the General Electric Company. Two halves of a large valve chest approximately 7 ft long with $2^1/_2$-in-thick walls were welded together, and then a large stop-valve casing was welded to each end of the chest. Also, a Kelcaloy transition piece was welded to the inlet of each stop valve, in order that the field weld connecting the main steam piping to the stop valves could be made between similar materials. At several steps in the making of these welds Zyglo and microscopic examinations were made and x rays taken. No defect in the weld metal was discovered in any of these examinations. Lime-coated electrodes were used for all this welding, and no preheat was used. The major problem in welding austenitic castings appears to be the difficulty of detecting discontinuities in the base metal which

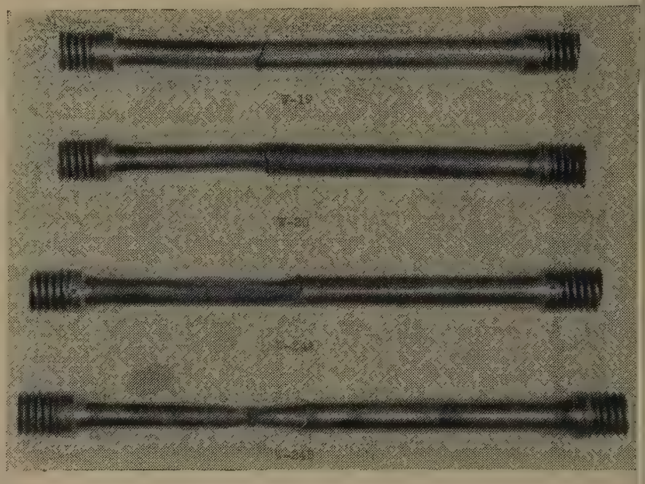

Fig. 21 Rupture Bars After Failure at 1100 F
(Bar W24B broke in less than 500 hr, the other three at times longer than 500 hr.)

Fig. 22 End View of Bars Shown in Fig. 22

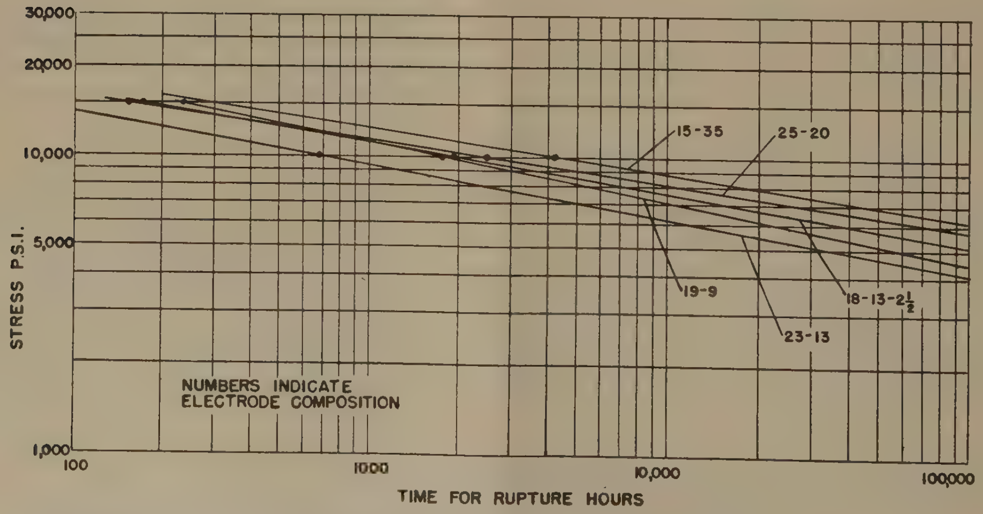

Fig. 23 Stress-Rupture Time Curves on Bars Cut From Multiweld Test Piece After Cyclic Heat Tests at 1100 F

may develop into large cracks under the influence of stresses set up by the welding.

As mentioned previously, test bars from each weld in the multiweld piece were placed in a rupture furnace and held at 1100 F until failure. The bars were loaded at various stresses to obtain data for a stress-time curve. All the bars loaded to stresses to cause failure in less than 500 hr failed in the $2^1/_4$ Cr-1 Mo at stress-time values comparable to those published for this material. However, even on these bars there was a sharp line of demarcation at the fusion zone where the ferritic material showed considerable strain and the austenitic material practically none. On bars loaded to stresses causing failure in times longer than 500 hr, an unusual type of failure showed up. Apparently, failure starts as a shearing at the fusion zone and then final failure occurs as a typical tension failure of the intercrystalline type with low ductility (see Figs. 21, 22). Bars from each weld are still running in the furnace, but the data obtained so far have been plotted (see Fig. 23), and the curves extrapolated to 100,000 hr to give the following rupture-stress values:

Electrode	100,000-hr rupture strength at 1100 F, psi
15 Cr-35 Ni	6200
19 Cr-9 Ni	4500
18 Cr-13 Ni-$2^1/_2$ Mo	5000
23 Cr-13 Ni	4100
25 Cr-20 Ni	5900

NOTE: All electrodes except the 15 Cr-35 Ni were stabilized with columbium.

Published data for the $2^1/_4$ Cr-1 Mo material give the following 100,000-hr rupture strengths at 1100 F:

B&W (annealed)	6300 psi
Timken (normalized)	8200 psi

At present a turbine to operate at 1050 F is being designed with stop valves of austenitic steel to which the main steam pipe of $2^1/_4$ Cr-1 Mo steel will be welded by a joint similar to the ones subjected to the tests described in the paper. The wall thickness has been proportioned to give safe stress values in accordance with test data.

I. A. ROHRIG.[10] All who have investigated austenitic welding of ferritic materials have been confronted with the same fundamental objection, namely, the effect of stresses due to the different expansion characteristics of dissimilar materials. The author has demonstrated that the expected high stresses either did not occur or were not severe enough to cause failure even under the conditions imposed by a quench. The published results of other investigators are in general agreement with those reported by the author.

Some of the earliest work on austenitic welding of ferritic materials appears to have been done in the Krupp Industries as judged by published reports. European investigators were exploring austenitic welding at that time for use in boiler construction. The theoretical objection was raised, however, that owing to the greater coefficient of thermal expansion of the austenitic weld metal as compared with that of the ferritic boiler-construction materials, there would be produced intolerably high differential stresses which would vary considerably with heating and cooling.

Kautz[11] stated that alternating hydrostatic-pressure tests on boiler drums containing austenitic welds, as well as tempering and cooling tests on austenitic-welded structures, did not substantiate the fear that the differential stresses between ferritic base material and austenitic weld metal were dangerous because of different thermal expansions. A further result of those earlier studies by both Beckman and Kautz was the Krupp "special weld," German Patent 567094, pertaining to the application of austenitic welding in boiler and auxiliary-equipment construction.[12] The invention covers a chromium-nickel austenitic welding electrode of the nominal composition, 25 chromium-20 nickel, and was recommended for the welding of ferritic steels.

In 1936 The Detroit Edison Company used austenitic chromium-nickel electrodes to weld a carbon-molybdenum test section of $5^1/_2$-in-ID and $^3/_8$-in. wall thickness into an 18-8 stainless-steel experimental steam line operating at 925 F and 380 psi pressure. The welds were made using 18-8 electrodes, not stabilized. One such austenitic welded joint was examined in 1943, after it had been in service for 20,000 hr carrying steam at 925 F. During its service life, the joint had been cooled to room temperature 42 times. No harmful effects resulted from either the carbide precipitation at the fusion zone of the weld or the mechanical stressing of the joint, caused by the unequal expansion of the austenitic weld metal and the ferritic pipe metal.

In addition to the fact that the austenitic weld was entirely satisfactory for further service, a significant finding of that particular investigation was that the austentitic weld metal had a pronounced effect in suppressing graphitization in the weld-heat-affected zone of the ferritic pipe material.

Subsequent investigation confirmed this finding and the results of the study were reported in 1944.[13] Although austenitic welding was suggested at that time as a means for control of graphitization in carbon-molybdenum piping, the same objection, namely, stresses due to differential expansion, was raised. This objection appears to have been disposed of by the results reported from tests as well as from actual service.

A macrograph of the cross section of the austenitic-welded joint that had been in service for 20,000 hr at 925 F is shown in Fig. 24 herewith. A micrograph of the joint is shown as Fig. 25. Although a considerable amount of carbide had migrated from the carbon-moly pipe to the fusion zone of the 18-8 weld and into the weld metal, the Charpy notched-bar strength of the fusion

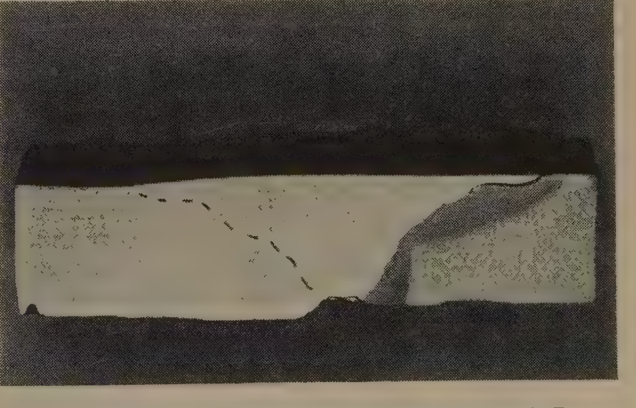

FIG. 24 CROSS SECTION OF AN 18/8 WELD BETWEEN 18/8 PIPE AT LEFT, AND C-MO PIPE AT RIGHT, AFTER 20,000 HR AT 925 F
(Black line at fusion zone between C-Mo pipe and 18/8 weld is composed of carbides and is not a crack; $\times 1^1/_2$ approx.)

[10] Research Department, The Detroit Edison Company, Detroit, Mich.

[11] "The Development of Austenitic Welding," by K. Kautz, *Technische Mitteilungen Krupp*, vol. 3, August, 1935, pp. 143–173.

[12] "The Krupp Special Weld: German Patent 567094," by E. Beckman, *Technische Mitteilungen Krupp*, vol. 3, August, 1935, pp., 137–142.

[13] "A Study of Austenitic Welding for Control of Graphitization in Steel," by I. A. Rohrig. See pamphlet on "Graphitization of Steel Piping," pp. 65–76, published September, 1945, by THE AMERICAN SOCIETY OF MECHANICAL ENGINEERS.

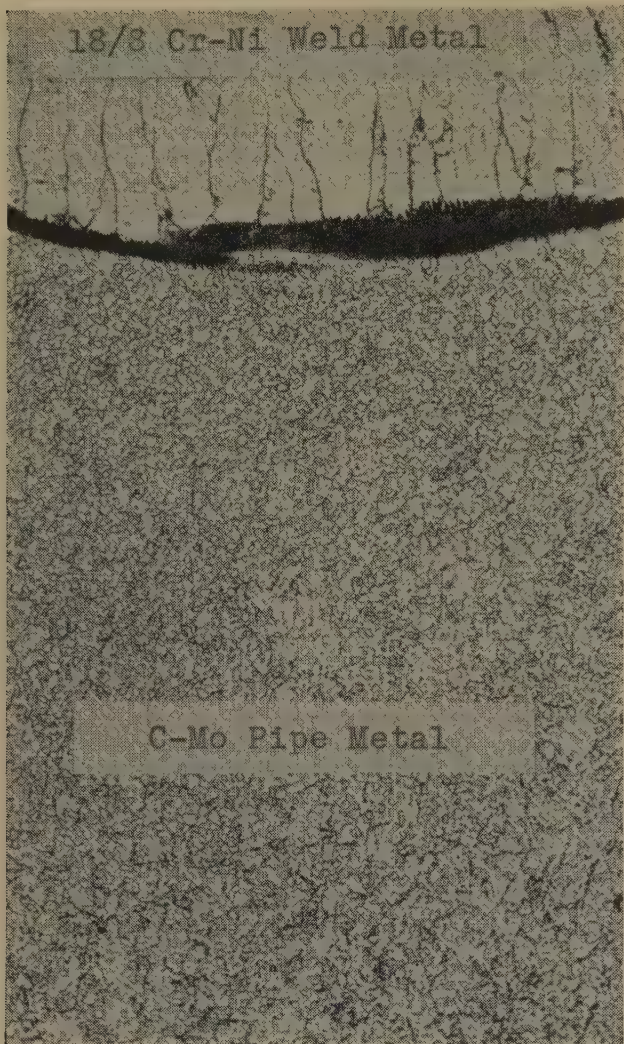

Fig. 25 Heat-Affected Area of 18/8 Cr-Ni Weld in C-Mo Pipe, Showing Carbide Band at Fusion Zone After 20,000 Hr of Service at 925 F; ×100

zone was excellent and showed no reduction as a result either of carbide precipitation and enrichment or of differential stressing due to heating and cooling in service. The notched-bar values obtained are given in Table 11 of this discussion. These results support those given in the paper.

Objections have been raised against austenitic welding of ferritic materials because of some failures in service. The failure may have been due to factors such as inadequate heat-treatment or improper choice of materials for the service involved. Austenitic welds, if made in carbon steel that is to be used at high temperature, may fail in the weld-heat-affected zone as a result of carbon depletion and grain growth in that zone during high-temperature service. As little alloy as 0.50 per cent molybdenum has been found to be effective in preventing such grain growth in the weld-heat-affected zone even though carbon depletion may still occur.

Austenitic welds likewise are not advisable for high-temperature applications in steels that develop a highly martensitic structure in the weld-heat-affected zone, unless the martensitic structure is fully tempered or annealed before service. The 1550 F annealing treatment used by the author would satisfy that requirement.

Fig. 16 of the paper shows a complete absence of a martensitic structure in the weld-heat-affected zone in the $2^1/_4$ chrome-moly base material, which is favorable. The structure indicates also that some carbon depletion has occurred in the weld-heat-affected zone. The alloy content of the base metal, however would prevent grain growth in that zone.

After examination of the test results reported for each of the different welds in the special test section used by the author, a question arises concerning his selection of the 19-9 welding electrode for future use. His test results have shown an absence of cracks in the 25-20 weld, whereas, when the 19-9 electrode was used, cracks developed. This fact, together with the lower expansivity and greater structural stability of the 25-20 weld, seems to favor the use of electrodes of that composition rather than of 19-9.

TABLE 11 NOTCH TOUGHNESS AT 70 F OF AN AUSTENITIC-WELDED JOINT AFTER APPROXIMATELY 20,000 HR AT 925 F

Location of notch in specimens	Toughness, ft-lb, Charpy V-notch
(a) In 18/8 Cr-Ni weld metal, at center of weld	60 70 78
(b) At junction between weld metal and pipe metal	110 105 190
(c) In pipe metal unaffected by weld	110 180

A. B. Wilder.[14] Stability at 1050 F (565 C) of the 18-8 Cb stainless steel and 3 Cr-1 Mo ferritic steel used in the Sewaren Generating Station over long periods of exposure is one of the factors which will have an important bearing on the ultimate properties of these materials. Data [15,16] developed in our laboratories on similar steels shown in Table 12 herewith, after exposure at 900-1050-1200 F (480, 565, and 650 C) for a period of 10,000 hr in the welded and unwelded conditions, will be discussed.

The prior heat-treatment before depositing weld beads and the welding electrodes used are shown in Table 13 of this discussion. The weld beads were deposited with a $^1/_8$-in-diam electrode using 100 amp at 24 volts, and an arc travel of 10 ipm. No preheating or postheating was employed.

Microstructure. Titanium carbides in the 18-8 Ti, and columbium carbides in the 18-8 Cb in the unexposed parent metal are

TABLE 12 CHEMICAL COMPOSITION OF STEELS

Code no.	C	Mn	P	S	S	Cr	Ni	Mo	Cb	Ti
24	0.07	0.52	0.021	0.003	0.386	17.97	10.40	0.011	..	0.58
25	0.06	1.43	0.018	0.006	0.444	17.80	11.16	0.076	0.77	.
52	0.11	0.45	0.011	0.021	0.390	3.21	0.21	0.97	..	.

TABLE 13 HEAT-TREATMENT AND WELDING ELECTRODES

Code no.	Steel designation	Heat-treatment prior to exposure	Welding electrode employed
24	18-8 Ti (321)	1900 F water quench 1550 F air cool	Type 347
25	18-8 Cb (321)	1550 F air cool, quench	Type 347
52	3 Cr-1 Mo	1550 F anneal	Type 502

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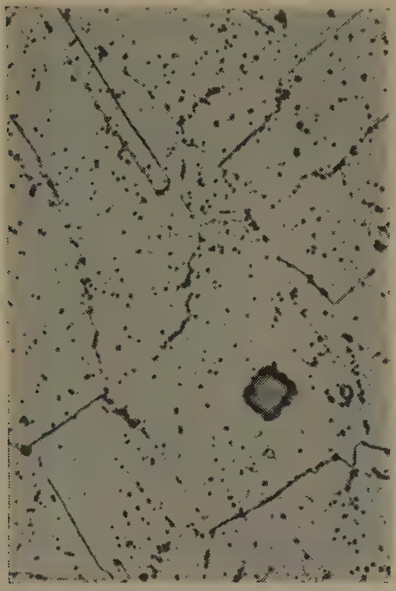
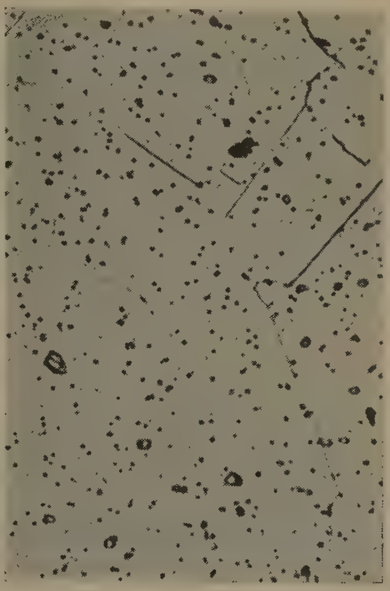
[14] Chief Metallurgist, National Tube Company, U. S. Steel Corporation of Delaware Subsidiary, Pittsburgh, Pa.

[15] "Graphitization of Steel at Elevated Temperatures," by A. B. Wilder and J. D. Tyson, Trans. ASME, vol. 40, 1948, p. 233.

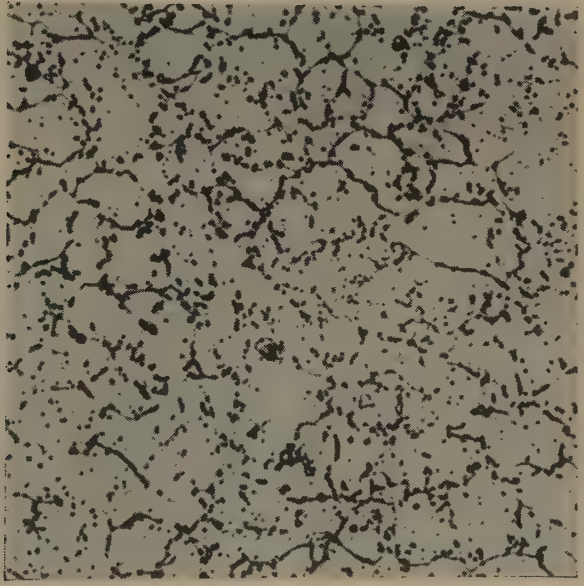
[16] Stability of Steels at Elevated Temperatures," by A. B. Wilder and J. O. Light, Preprint no. 36, American Society for Metals, 1948.



(a)



(b)



(c)

FIG. 26 UNEXPOSED PARENT METAL; ×1000
(a, 18-8 Ti oxalic-acid etch, showing titanium carbides; b, 18-8 Cb oxalic-acid etch showing columbium carbides; c, 3 Cr-1 Mo nital etch showing carbides.)

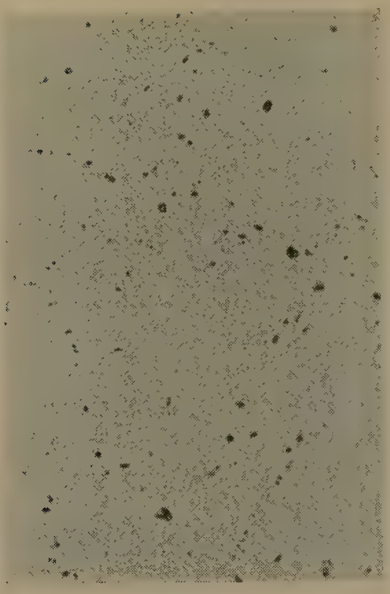
shown in Fig. 26 *a*, *b*, and *c* herewith. Grain structure of the weld heat-affected zone of these steels before exposure had a tendency to have a columnar structure normal to the weld metal junction.

In the 18-8 Ti and 18-8 Cb parent metal, sigma phase was observed after 10,000-hr exposure at 1200 F (650 C). The occurrence of this phase was questionable at 1050 F (565 C) and was not observed at 900 F (480 C). Sigma phase in the 18-8 Ti steel is illustrated by comparison of Figs 27(*a*) and (*b*). Sigma phase was also observed in the weld-heat-affected zone of the 18-8 Ti and 18-8 Cb steel after 10,000-hr exposure at 1050 and 1200 F (565 and 650 C). A larger quantity of the constituent was present in the weld-heat-affected zone than in the parent metals.

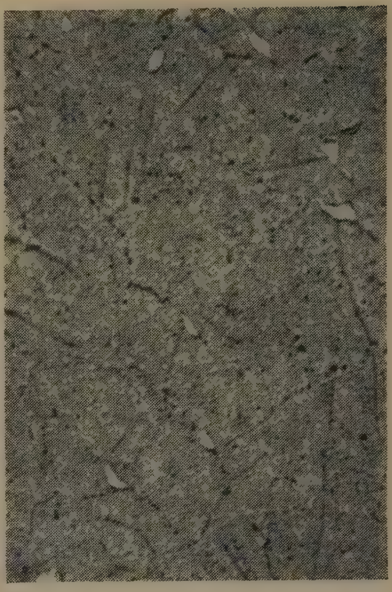
In the 3 Cr-1 Mo, no graphitization was observed after exposure for 10,000 hr at 900, 1050, and 1200 F (480, 565, and 650 C). No change in microstructure was detected after exposure at 900 F. A slight agglomeration of carbides occurred at 1050 F after exposure, and at 1200 F additional agglomeration was observed.

Impact and Hardness Properties. Impact properties of the several steels are shown in Table 14. The lower impact properties of the 18-8 Cb and 18-8 Ti steels after 10,000 hr exposure at 1200 F (650 C) were associated with the presence of sigma phase. No explanation is offered at this time for the lower impact properties of the 3 Cr-1 Mo steel after exposure.

Hardness changes are presented in Table 15 of this discussion. The austenitic steels became slightly harder at some exposure



(*a*)



(*b*)

Fig. 27 Parent Metal After Exposure of 10,000 Hours at 1200 F; ×1000

(*a*, 18-8 Ti Murakami's reagent showing carbides; *b*, same field as *a* after repolishing and etching with Vilella's reagent showing carbides and white constituent sigma phase.)

TABLE 14 IMPACT PROPERTIES

Steel designation	Charpy full-size keyhole notch specimens, ft-lb Before exposure	Exposed 10000 hr 900 F	1050 F	1200 F
18-8 Ti (321)	107	88	72	62
18-8 Cb (347)	56	63	51	32
3 Cr-1 Mo	77	52	39	41

temperatures. These changes were probably associated with a precipitate, submicroscopic in size. The slight change in hardness of the 3 Cr-1 Mo steel after exposure at 1200 F may be associated with carbide agglomeration.

TABLE 15 HARDNESS PROPERTIES

Steel designation	Rockwell B hardness Before exposure	exposed 10,000 hr 900 F	1050 F	1200 F
18-8 Ti (321)	81	82	83	89
18-8 Cb (347)	88	81	85	84
3 Cr-1 Mo	77	75	74	71

The material described in this discussion will be exposed for a period of 10 years. Creep resistance, tensile properties, and other mechanical, as well as certain chemical properties, will be determined after exposure.

N. L. Mochel.[17] It was indeed a pleasure to co-operate with the author and his associates of the Public Service Electric and Gas Company, and with the representatives of the General Electric Company, the M. W. Kellogg Company, the Combustion Engineering Company, and the United Engineers and Constructors, Inc., in the planning and carrying out of the tests described so well in this paper.

Along with others, the writer's company received a long section of the welded composite after all of the tests were completed, and some studies made on this specimen may be of interest in connection with a study of the entire matter.

A longitudinal specimen from the Kelcaloy test piece was secured, and this was cut up to observe the general nature of the joint of the cast to the wrought material. Visual and microscopic examination and bend tests left the impression that the cast 18/8 stainless steel and the wrought chromium-molybdenum steel were well bonded together, and no separation had taken place during the cycling tests.

A section $2^1/_2$ in. wide for the full length of the multiweld test piece, Figs. 1 and 2 of the paper, was procured. This piece was identified as J-10. The five welds in the longitudinal section were identified in order as follows:

Designation	Deposited metal	
A	15 Cr–35 Ni	
B	19 Cr– 9 Ni Cb	Buttered with 25/20
C	18 Cr–13 Ni–$2^1/_2$ Mo–Cb	Buttered with 25/20
D	23 Cr–13 Ni–Cb	
E	25 Cr–20 Ni	

The full length of specimen was first ground, polished, and etched to show the general structure. The results are recorded in Figs. 28, 29, and 30 of this discussion, which should be grouped together to show the full length. Close examination shows small root cracks in the case of welds A and B. The other three were sound.

Figs. 31 and 32, herewith, show the manner of cutting up this longitudinal specimen to provide creep-rupture and creep specimens. Four such specimens were prepared from each welded joint. Creep and creep-rupture tests were made on most of these specimens and gave results as shown in Table 16 of this discussion.

TABLE 16 RESULTS OF CREEP AND CREEP-RUPTURE TESTS

Specimen	Test load, psi	Results of tests
A-1	40000	Failed in 3 hr; failure in Cr-Mo steel
A-2	20000	Discontinued after 158 hr; 14.4 per cent elongation
A-3	8000	Discontinued after 500 hr
B-1	8000	Still running after 7800 hr; present creep rate is 4 × 10^{-8} per hr
B-2	12000	Failed after 3400 hr in joint of weld to Cr-Mo steel
B-4	20000	Failed after 270 hr, with 32 per cent elongation in mid-length of Cr-Mo steel
C-1	40000	Failed after $2^1/_2$ hr in CrMo steel
C-2	20000	Failed after 139 hr, with 20.4 per cent elongation in CrMo steel
C-3	8000	Discontinued after 500 hr
D-1	8000	Still running after 7800 hr. Present creep rate is 5.3 × 10^{-8} per hr
D-2	12000	Failed after 5420 hr in fillet of Cr-Mo steel
D-3	20000	Failed after 105 hr in joint of weld to Cr-Mo steel
E-3	12000	Discontinued after 3000 hr; 1.88 per cent elongation
E-4	8000	Discontinued after 3000 hr; 0.28 per cent elongation

[17] Manager, Metallurgical Engineering, Westinghouse Electric Corporation, East Pittsburgh, Pa.

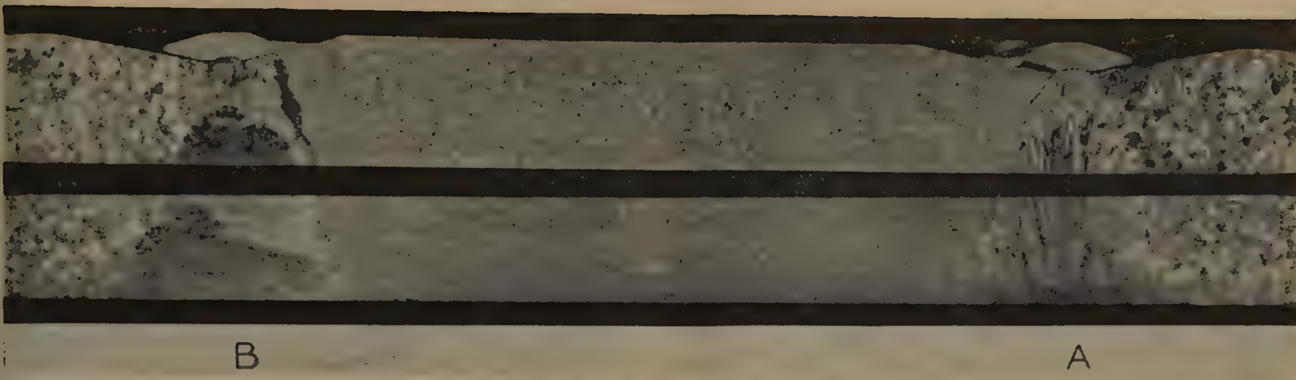


Fig. 28

Fig. 29

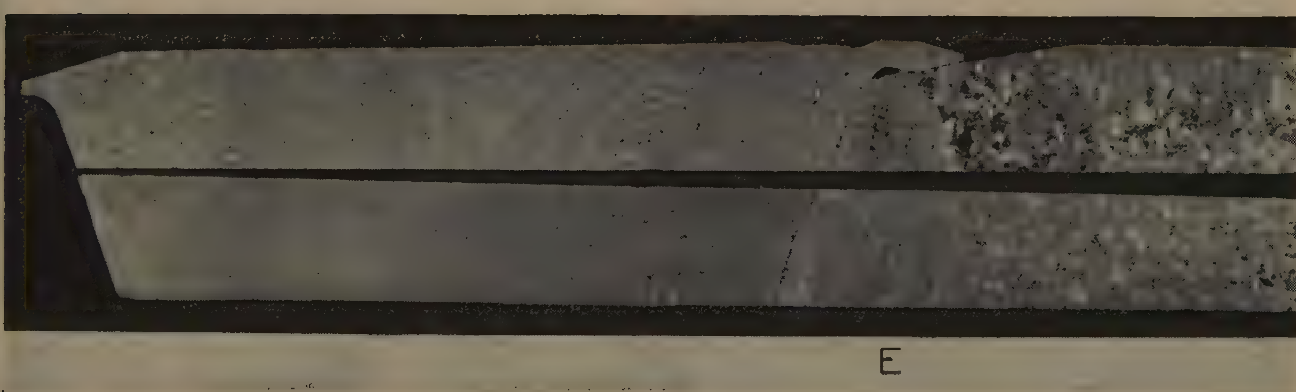
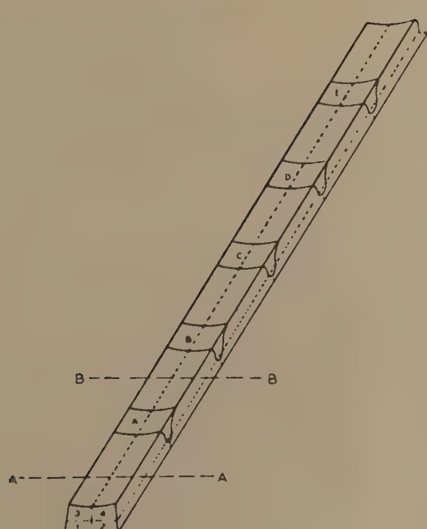


Fig. 30

Fig. 31 *(left)*



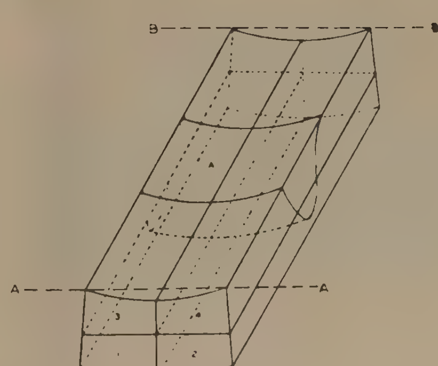


Fig. 32

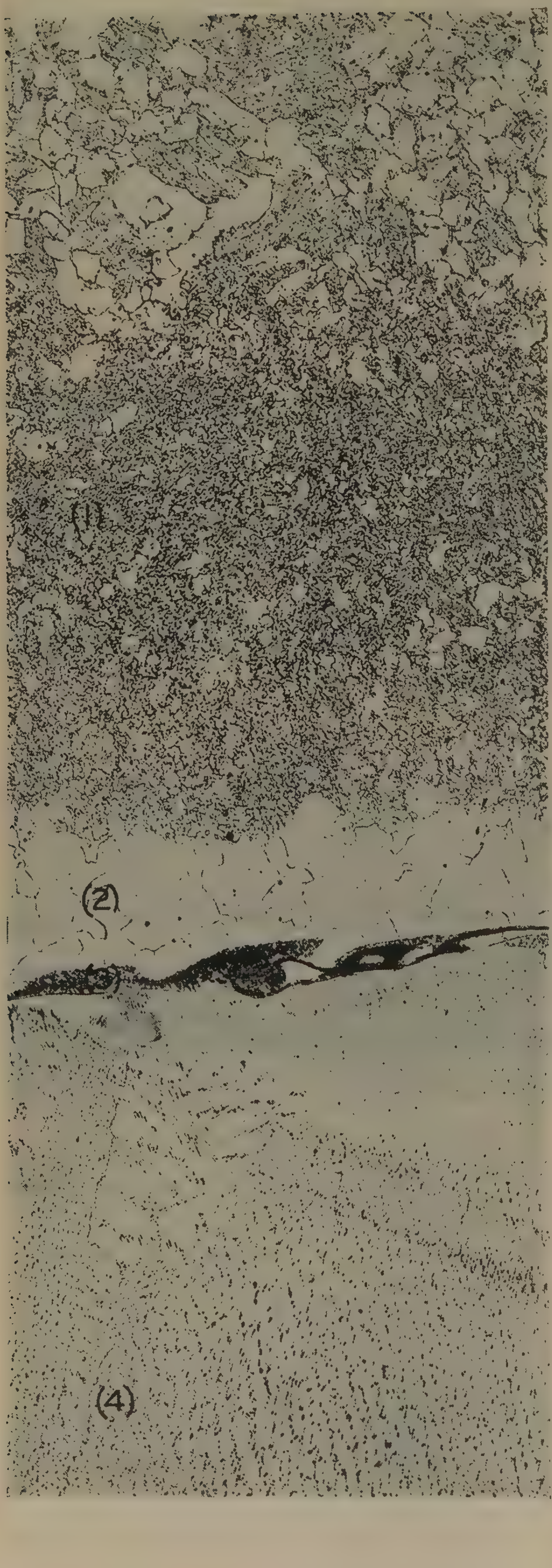


Fig. 33 Microstructure of 18–8 Ferrite Welded Joint

(1) Ferrite material
(2) Decarborized zone
(3) Carbide migration
(4) Buttered 25–20
(5) Heat-affected zone
(6) Weld 18–13–2–$^1/_2$
(7) Cast 18–8

Etchant: (1)–(2) and (3) 2 per cent Nital; (4)–(5) and (6) and (7) 10 per cent Chromic Electrolytic

Magnification: ×60

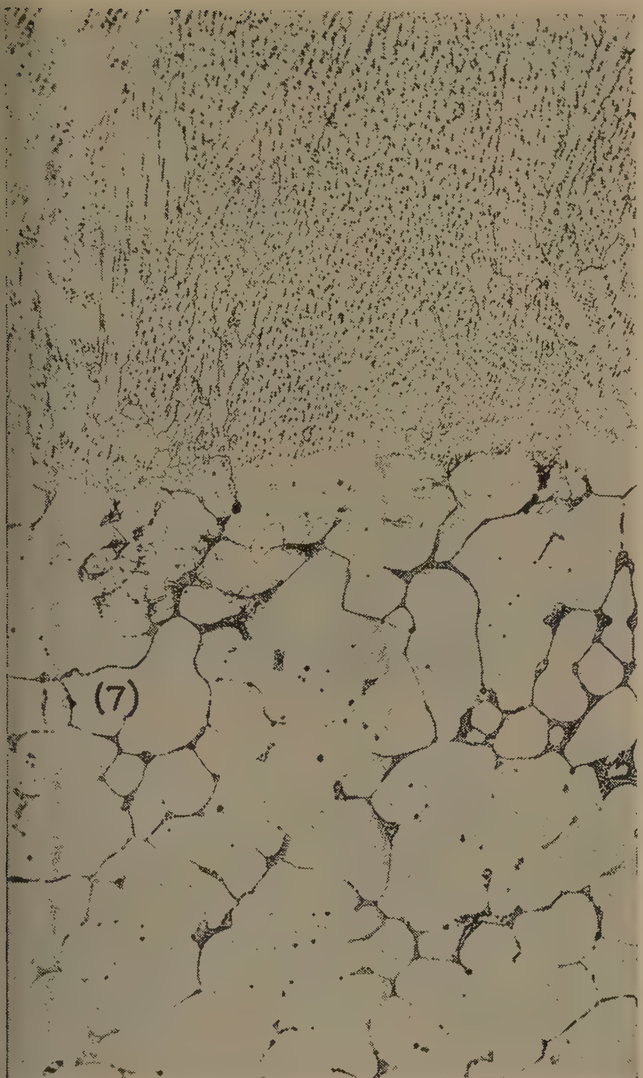


Fig. 33 (*Continued*)

Fig. 33 shows a composite micrograph of the C-weld. It may be of interest especially because it shows the decarburized zone where the weld joins the ferritic material. This matter of carbide migration will receive much comment and thought. It is of interest that some of the specimens mentioned failed at this point in the creep-rupture tests. This matter will bear further examination.

Author's Closure

Messrs. Blumberg's, Jackson's, and Mochel's discussions should be considered as supplementing the paper, much of the work involved in the test having been carried out by them. Mr. Conrad's stress analysis also helps make the presentation more complete.

Mr. Akin's tests on tubing are helpful in evaluating the various heat-treatments which may be used. Because his welds were so much smaller, he was able to cover the entire range of possibilities. This preliminary work was not practical with the large pipe used in our test. His specimens 4A-6 and 4A-7 (Table 6) which were postheated to 1550 F and showed elongations of only 5.0 and 2.5 per cent, respectively, would appear to explain the low ductility of the multiweld piece of our test. Messrs. Blumberg and Brister point out, and our independent observations of other work indicate, that this heat-treatment does lower ductility. The resultant structure with 1550 F postheat, however, may represent more closely that which would be expected in actual service after several years of operation at 1050 F, and therefore may have been more desirable for the purpose of this test.

Mr. Akin's evaluation is based entirely on ductility. As Mr. Robinson points out, ductility tests may not be appropriate for judging the suitability of materials for long-time service at extreme temperatures.

Mr. Akin's suggestion that a 25-20 transition piece be used to make the joint between the dissimilar steels may have considerable merit for superheater tubing, where a large number of joints must be made under confined space conditions, and therefore the elimination of the field postheating would result in considerable saving in erection costs.

In reply to Mr. Akin's specific questions, the multiweld piece was stress-relieved before it was allowed to cool from the preheat temperature. As reported in the discussion by Messrs. Blumberg and Bunn, the Kelcaloy piece "was annealed by heating to 1550 F, holding for 4 hr, followed by furnace-cooling."

Referring to Mr. Rohrig's comment on the choice of 19-9 welding electrodes for future use in making a weld between the dissimilar materials, this filler metal was selected tentatively because of the desire to have the weld as nearly the same composition as the base metal as possible. The better weld obtained with the 25-20, compared to the 19-9 in the multiweld piece of the test, should not be taken as being representative of the two filler metals, as these were the first welds made, and much has been learned regarding welding heavy austenitic sections since that time. More information is expected to be available before it will be necessary to make our decision.

Mr. Wilder's investigation of the stability of various steels at operating temperatures will be followed with great interest. There is no complete substitute for time in determining the behavior of metal at high temperature.

The wide interest in the subject of this paper, shown by the large amount of discussion, is gratifying. It is evident that welding high-pressure austenitic piping is an involved problem subject to many contradictory ideas. Because of the high coefficient of expansion, low conductivity, and low yield point of austenitic steels, the welding of heavy sections is particularly difficult. When we superpose on all this the problem of welding dissimilar materials, with the possibility that what is good for one is not good for the other, we have a real job on our hands.

The following are some of the factors which have been suggested as being important in the welding of austenitic steels:

Carbon content	Castings or forgings
Silicon content	Preheat temperature
Cb or Ti content	Postheat temperature
Cb — C ratio	Peening
Si — C ratio	Buttering
Cb — Si ratio	Type of weld bead
Cr — Ni ratio	Welding current
Delta-ferrite orientation	Weld-rod coating
Grain structure	Backing rings

Room-temperature physical properties of test coupons are the criterion by which the importance of the various factors are judged.

The end result desired is a piping system which will stand up satisfactorily in power-plant service at the specified temperature for as long a time as possible, say, up to 40 years. Room-temperature physical properties of test coupons serve only as an indication of the soundness of the joint. They are no measure of the high-temperature properties of the full-size joint, nor of the stability of the metal at high temperature. High-temperature testing, either full scale or of test coupons, is costly and time-consuming,

but this is what is needed if we are to avoid failures in service. Admittedly, these cyclic heating and cooling tests just scratch the surface.

It is comforting to know that even though the welds in the multiweld piece were not of good quality, as judged by the customary but, nevertheless, arbitrary tests of room-temperature physical properties, no apparent change resulted from cyclic heating and cooling. The use of the Kelcaloy joint appears to pr vide an additional factor of safety, and therefore in our opinio is justified in view of the uncertainties of the situation.

At this date the first two units at Sewaren have been in oper tion about 5 months. Plans for the additional tests mentioned the paper are well advanced, and it is hoped that more i formation will be available soon.

Energy in the Engine Exhaust

By P. H. Schweitzer[1] and T. C. Tsu,[2] State College, Pa.

The maximum amount of energy which an ideal turbine can recover from the exhaust gas of an engine is the same as the additional energy obtainable from an ideal, complete-expansion piston engine. This energy, known as the convertible blowdown energy, is equal to the tail triangle of the *p-v* diagram. A chart is presented for the quick determination of the convertible blowdown energy for exhaust release pressures between 10 and 1000 psia, exhaust release temperatures between 0 and 4000 F, and altitudes between sea level and 40,000 ft.

In view of the fact that 30 to 50 per cent of the energy of the fuel released during combustion in an engine escapes through the exhaust, it is not surprising that engineers pay more and more attention to the recovery of the exhaust energy. That generally takes one of two forms. The energy of the exhaust is either used for heating, or it is used for producing mechanical work. The thermal utilization of the exhaust, which pertains to heat exchangers and exhaust boilers, is outside of the concern of this paper.

Exhaust Energy for Power Production

Recently the exhaust energy is being utilized more and more for the production of power. Compound engines, exhaust turbines, and inertia charging of cylinders belong to this category. Exhaust turbines are now widely used both for aircraft and Diesel engines for power boost. Ordinarily the power gain is only indirect. The exhaust turbine drives a blower which supercharges or helps to supercharge the engine. In some applications, however, the turbine has been geared to the main shaft to augment the engine power.

While the turbine is the most popular means of utilizing the exhaust energy, it is helpful to know that the maximum amount of work which an ideal turbine is capable of producing is exactly equal to the work an ideal engine would deliver if the same exhaust gas were allowed to expand in both cases to the same ambient pressure. One of the authors called attention to this fact[3] in 1925 but it has been largely ignored.

"Complete-Expansion" Piston Engine. In case the exhaust energy is utilized in a second, complete-expansion, piston engine, the gain is evidently equal to the tail triangle of the *p-v* diagram, shown shaded in Fig. 1. Line 4–1 is the constant-volume line at the bottom center position of the piston from exhaust release pressure to ambient pressure. Line 4–5 is an isentropic from point 4 to ambient pressure. Line 1–5 represents the ambient pressure, which at sea level is 1 atm. The area of the triangle can be expressed either as

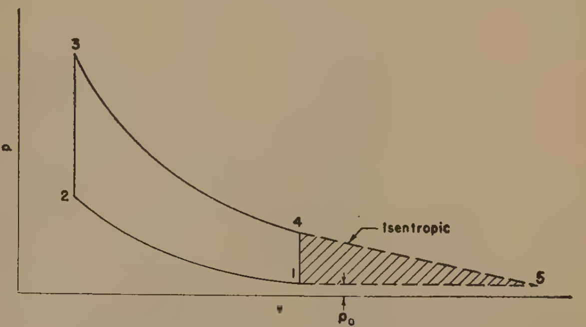


Fig. 1 Convertible Blowdown Energy—Shaded Triangle

Convertible blowdown energy $= W_e = (u - u_0) - Ap_0(v_0 - v)$ or as

$$W_e = (h - h_0) - A(p - p_0)v$$

where u and u_0 represent the internal energy of the gas at points 4 and 5, h and h_0 the enthalpy at those points, and A the reciprocal of the mechanical equivalent of heat. In using the latter formula and assuming the exhaust to be a perfect gas

$$W_e = (h - h_0) - ART\left(1 - \frac{p_0}{p}\right)$$

Exhaust Turbine. In the other case the exhaust energy is recovered by an exhaust turbine during the blowdown process. Assuming an ideal turbine which utilizes 100 per cent of the isentropic enthalpy drop across the turbine all the time, while the gas blows out of the cylinder until the cylinder pressure becomes ambient atmospheric, the total turbine work is

$$W_t = \int (h_x - h_0)dG$$

where h_x is the instantaneous enthalpy, and G the mass of the gas in the cylinder. Envisaging 1 lb of gas before exhaust release, the integration limits are 1 and the final amount of gas G_0 which remains in the cylinder when the blowdown process is completed.

It can be shown mathematically or by a heat balance that

$$\int_{G_0}^{1} (h_x - h_0)\,dG = (h - h_0) - ART\left(1 - \frac{p_0}{p}\right)$$

In a particular case p = 180 psia, T = 3300 R and p_0 = 14.7 psia

$$(h - h_0) - ART\left(1 - \frac{p_0}{p}\right) = (784.19 - 356.69) - \frac{53.3 \times 3300}{778}\left(1 - \frac{14.7}{180}\right)$$

$$= 220.5 \text{ Btu per lb}$$

$$\int_{G_0}^{1} (h_x - h_0)\,dG$$

is also evaluated in Fig. 2 by graphical integration, always

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[1] Professor of Engineering, Research and Consulting Engineer, The Pennsylvania State College. Mem. ASME.

[2] Research Associate, The Pennsylvania State College. Mem. ASME.

[3] "How Much Can Be Gained by Exhaust Turbines?" by P. H. Schweitzer, *Mechanical Engineering*, vol. 48, 1925, pp. 860–861.

Contributed by the Oil and Gas Power Division and presented at the Annual Meeting, New York, N. Y., November 28–December 3, 1948, of The American Society of Mechanical Engineers.

Note: Statements and opinions advanced in papers are to be understood as individual expressions of their authors and not those of the Society. Paper No. 48—A-56.

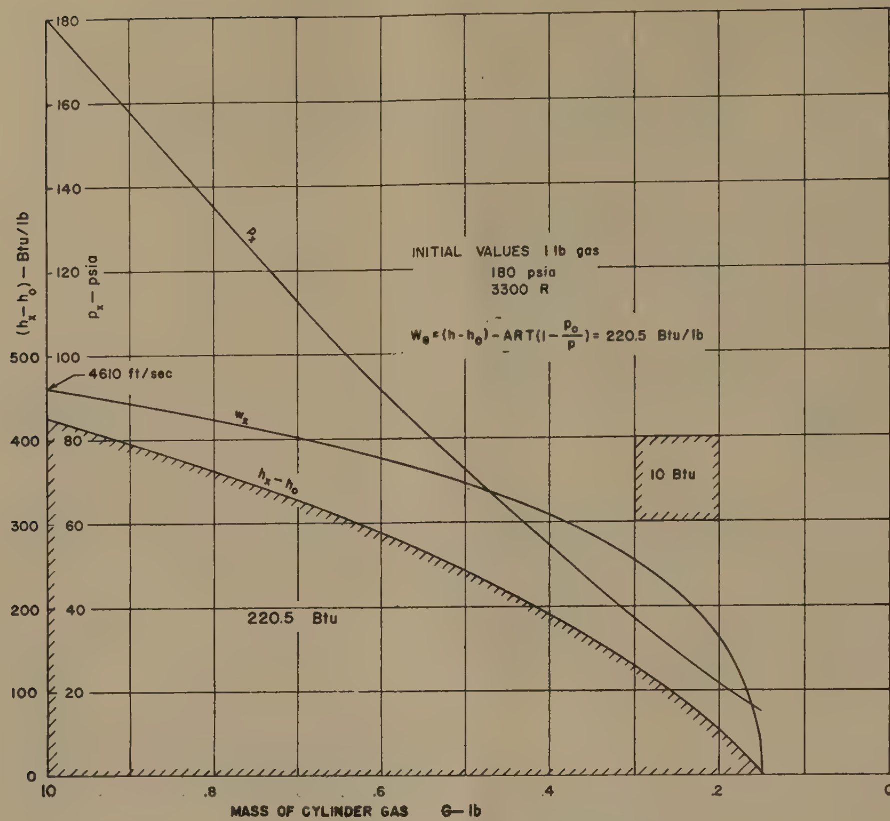


Fig. 2 Curves of p_x, w_x, and (h_x-h_0) Versus G

assuming $G_x = p_xV/RT_x$ and using Keenan and Kaye's[4] tables in the computations. The planimetration of the area under the (h_x — h_0) line gives exactly 220.5 Btu per lb and serves as an empirical proof (if one is needed) that the ideal engine delivers the same amount of work as the ideal turbine.

Fig. 2 illustrates the variation of the cylinder pressure p_x, the isentropic enthalpy drop (h_x — h_0), and the theoretical discharge velocity w_x, as the mass of cylinder gas decreases during the blowdown process. It may be noted that after the completion of the blowdown process, about 15 per cent of the gas mass remains in the cylinder.

Steady-Flow Turbine. If the same pound of cylinder gas were used in connection with a steady-flow turbine (the steady-flow condition being obtained by interposing an infinitely large receiver between the turbine and an assumed eight-cylinder engine), the maximum ideal turbine work would be only 129 Btu,[5] or 58.5 per cent of the convertible blowdown energy.

Convertible Blowdown Energy

It is clear that in order to obtain the maximum amount of energy from the blowdown gas, each gas element should be utilized at its highest possible pressure and temperature. During the early part of the blowdown process, the pressure and temperature of the discharging gas are high. Naturally, more available energy can be obtained from this high-pressure high-temperature gas than if it were allowed to mix with the later portions of exhaust gas, thereby reducing its pressure and temperature. The convertible blowdown energy, therefore, is the theoretical maximum amount of energy which can be recovered from the blowdown gas. It can be regarded as the goal which designers of exhaust turbines may try to approach, but never quite reach.

This fact gives the concept "convertible blowdown energy" a more than casual significance. It represents the maximum theoretical amount of energy that can be recovered from 1 lb of cylinder gas if it is allowed to expand to ambient atmospheric pressure. The "convertible blowdown energy" is defined by the initial condition of the gas p and T and by the ambient pressure to which the gas can expand.

It should be noted that in the foregoing formulas, W_e is the convertible blowdown energy expressed in Btu per pound of gas that exists in the cylinder at the point of exhaust release. For a given engine the total convertible blowdown energy is

$$U = W_e w_g \text{ Btu per min}$$

where w_g = the mass rate at which cylinder gas is produced (lb per min).

$$w_g = (n)\left(\frac{\pi}{4}D^2\right)(S)\left(\frac{1}{r-1}+\frac{S_e}{S}\right)\left(\frac{N}{c}\right)\left(\frac{p}{RT}\right)$$

where

w_g = lb of cylinder gas produced per min
n = number of engine cylinders
D = cylinder bore, in.
S = piston stroke, in.
S_e = effective expansion stroke = piston stroke at point of exhaust release, in.
r = compression ratio
N = engine speed, rpm
c = constant = 1 for two-stroke-cycle engines, or 2 for four-stroke-cycle engines

[4] "Thermodynamic Properties of Air," by J. H. Keenan and J. K. Kaye, John Wiley & Sons, Inc., New York, N. Y., 1945.

[5] Based upon yet unpublished data by one of the authors.

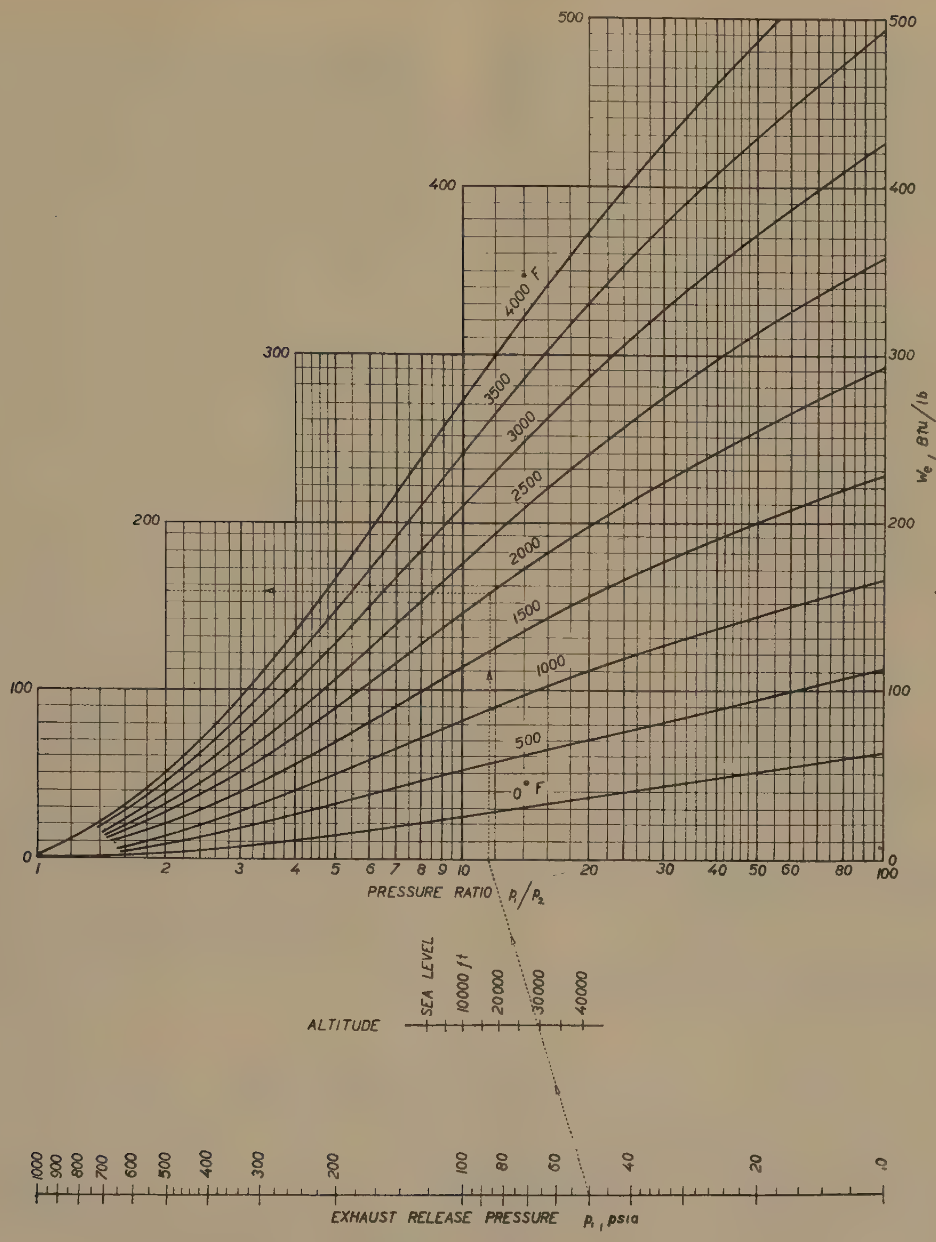


FIG. 3 CHART FOR DETERMINING CONVERTIBLE BLOWDOWN ENERGY

(Example: At 30,000 ft altitude, ambient pressure = 4.36 psia; exhaust release temperature = 2000 F; exhaust release pressure = 50 psia; pressure ratio = 11.5; convertible blowdown energy = 156 Btu per lb of cylinder gas.)

p = exhaust release pressure, psia
T = exhaust release temperature, deg R
R = gas constant = 53.3 × 12 if cylinder gas is assumed to have properties of air

REFERENCE CHARTS

Charts shown in Fig. 3 have been prepared on the basis of air tables for ready reference in computing the convertible energy contained in the exhaust. It shows the convertible portion of the heat energy in 1 lb of cylinder gas, from sea level to 40,000 ft altitude.

In order to cover a wide range of pressure ratios, the abscissa is in logarithmic scale. From the exhaust release pressure and the altitude, the pressure ratio is obtained by an alignment chart below the abscissa axis.

The example shows that at 30,000 ft altitude with an exhaust of 50 psia release pressure and 2000 F release temperature, the convertible blowdown energy is 156 Btu per lb of cylinder gas. Compared with the fuel energy which approximately amounts to 1350 Btu per lb of air in a spark-ignition engine, the convertible blowdown energy is some 11.5 per cent. At sea level the same exhaust has only 60 Btu blowdown energy available to boost the thermal efficiency by 4.5 per cent if full recovery is accomplished. This means that, even theoretically, of the energy which escapes through the exhaust, only about one fifth can be recovered. At high altitudes a larger recovery is obtainable.

The chart is based on the thermodynamic properties of air. In order to show that with combustion gases the results are only slightly different, in Fig. 4 the convertible blowdown energy of pure air and that of the combustion products of a chemically

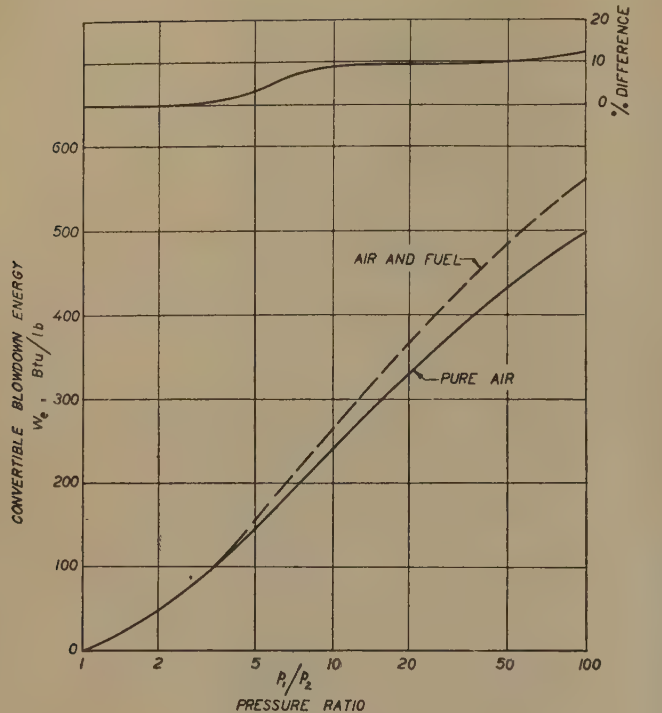


Fig. 4 Effect of Fuel

(Assumptions: T_1 = 4000 R; p_2 = 14.70 psia; fuel = $[CH_{2.25}]_x$; fuel/air = 0.0665.)

correct fuel-air mixture are compared. In the computation for the curve marked "air and fuel," a chart of thermodynamic characteristics prepared by Hershey, Eberhardt, and Hottel was used.

In view of the fact that the exhaust release pressure is seldom more than 6 times atmospheric, the error committed by using the air chart is negligible.

Discussion

Israel Katz.[6] The authors have added materially to the advance of internal-combustion engines and have presented a convincing case favorable to piston-engine compounding. The following remarks are made solely to illustrate how energy in the engine exhaust may perhaps best be employed in aircraft propulsion machinery.

While utilization of exhaust energy affords a ready means for boosting the power output of many engines, it is quite evident that aircraft piston engines have reached a point in development where substantial improvements yield only marginal performance advantages. Turbosuperchargers and coupled gas turbines offer but minor gains in net propulsive effort over the thrust boosts attainable with relatively simple duct-type augmenters or jet stacks (devices which involve little penalty in terms of mechanical complexity, increased weight, operating hazards, costs, and installation difficulties).

However, a 15 per cent power boost for sea-level take-off and moderate altitude cruise (up to 20,000 ft) without loss in fuel economy currently justifies the complexities of turbine compounding, and the leading engine builders have recently announced their contributions to this art. On the whole, such gains are only of temporary importance (long-range heavy bomber propulsion) and involve engine and airframe design changes which already crowd the point of marginal return. Further reduction in the weight/power ratios of existing engines may involve serious losses to reliability or endurance. Higher compression ratios or operation at very high manifold pressures will now yield only minor gains in economy or power output within limits set by incidence of detonation, spark-plug fouling, lead deposition, bromide attack on combustion-space components, increased fuel-octane requirements, and inadequate cylinder cooling. Operation at higher piston speeds will contribute to some power advantages, but also to greater mechanical losses, deleterious vibration, piston scuffing, and abnormal bearing wear.

[6] Associate Professor, Cornell University, Ithaca, N. Y. Jun. ASME.

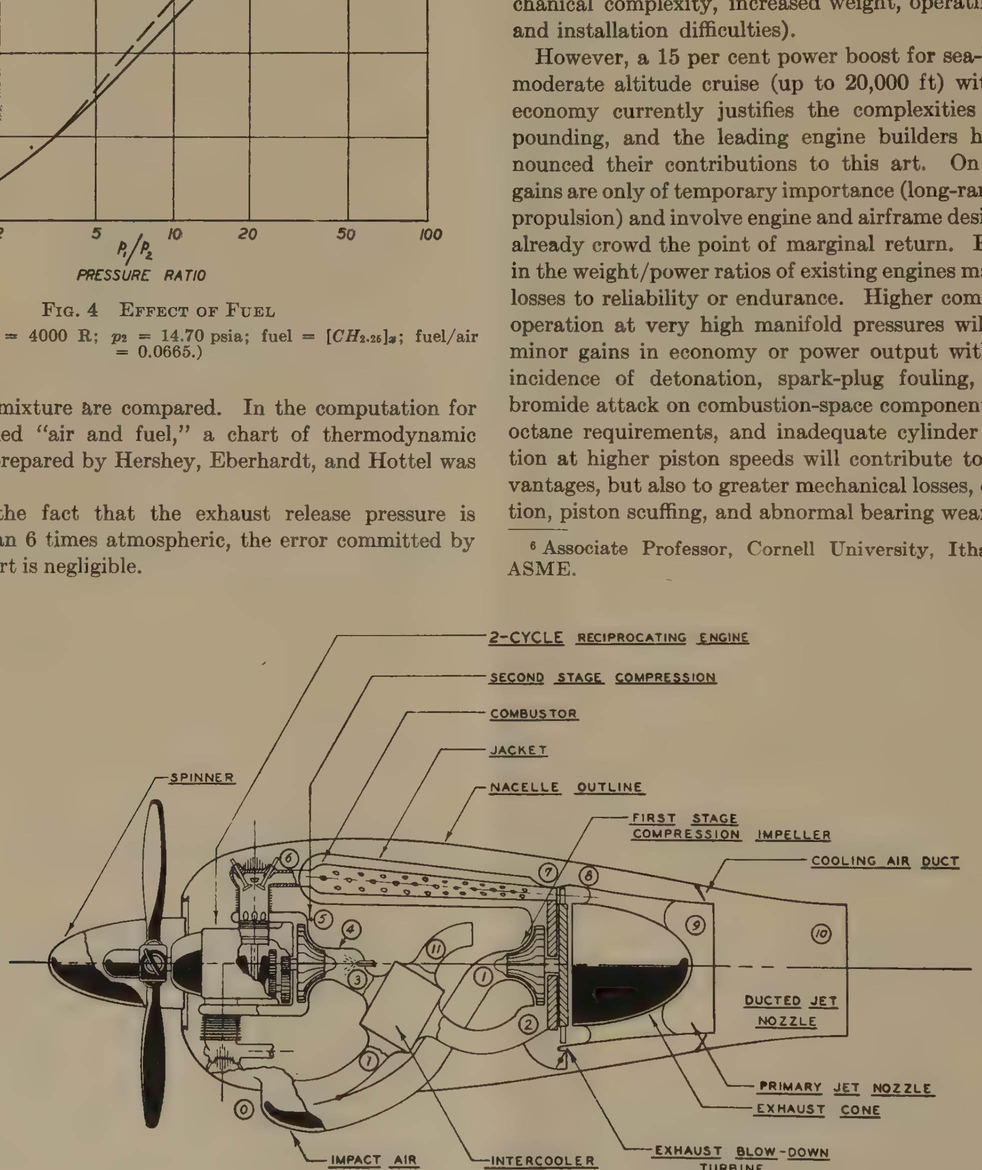


Fig. 5 Turbo Topping Power-Plant Installation Composed of a Mechanically Supercharged Two-Stroke-Cycle "Topping" Engine, Several Combustors, a Turbosupercharger and Ducted Jet Nozzle

(Air passing through first-stage compressor supplies engine and combustors. Engine exhaust gases 6, combine with air from the first-stage compressor 2, expand through the turbine 7, and primary jet nozzle 9, and finally merge with cooling air to provide substantial thrust boost 10. Drawing by Lieut. Col. E. E. Ambrose, U.S.A.F., working on this development under direction of discusser at Cornell Aircraft Power Plants Laboratory.)

Perhaps a novel approach to this seemingly exhausted endeavor may expose a new field for further developments. Highly refined engines surely merit the attention that a new viewpoint can bring to bear.

A recent classroom study, at the Sibley School of Mechanical Engineering, devoted to significant changes in aircraft-engine design and compounding possibilities, tentatively discloses that power-plant performance may be substantially improved if piston engines were used as "topping units" in combination with turbo-power-plant components. The power-plant installation evolved from this study (Fig. 5 of this discussion) may have some very promising features, such as:

1 High power output at moderate crankshaft speeds to be achieved through two-stroke-cycle operation, substantial supercharge, and uniflow scavenging during extended valve overlap. The use of very rich fuel-air mixtures should permit operation at high charge densities with low-octane fuels and afford a measure of internal cooling during the scavenge period.

2 Unusual over-all fuel economy (for the installation weight/power ratio) to be made possible by afterburning prior to turbine expansion (consistent with maximum allowable turbine-blade temperatures), in addition to the use of exhaust energy for supercharger operation and thrust augmentation in a ducted jet nozzle (where combustion products are to merge with heated cooling air).

3 Low weight/maximum-power ratio to be attained through two-stroke-cycle engine simplicity and the use of relatively light auxiliary components.

4 Low cost/power ratio to be achieved through use of existing highly developed piston engine, turbosupercharger and gas-turbine components.

5 Good installation versatility and maintenance ease.

Design details cannot be discussed, but the following information will illustrate other anticipated installation properties:

1	Displacement (piston engine), cu in	2000
2	Installation weight, lb	2900
3	Sea-level take-off power, bhp	3600
4	Static jet thrust (take-off), lb	800
5	Take-off bsfc (installation), lb per bhp-hr	0.56
6	Take-off crankshaft speed, rpm	3000
7	Take-off bte, per cent	21.6
8	Manifold air pressure (TO), in. Hg abs	80
9	Release pressure (TO), in. Hg abs	60
10	Combustor pressure (TO), in. Hg abs	49
11	Combustor temperature (TO), deg F	1400
12	Cruise power at 10,000 ft and 400 mph, bhp	1800
13	Cruise jet-thrust at 10,000 ft and 400 mph, lb	600
14	Cruise bsfc (installation), lb per bhp-hr	0.43
15	Cruise power bte, per cent	28.2

E. C. Magdeburger.[7] The paper presents a simple and quick way of estimating the heat content or potential energy of the exhaust gases of an internal-combustion engine for any possible pressure and temperature of these gases and for any pressure or altitude of the ambient air. "To measure is to economize," is the sage observation attributed to Pascal. Hence this new knowledge of the amount of energy wasted in the exhaust will keep on pointing its accusing finger at the engineer and challenge him to find ways and means of preventing this waste in a world which is becoming more and more sensitive to the status of its rapidly dwindling petroleum reserves.

However, it is not enough, for instance, to know with reasonable certainty just how much petroleum is stored in the vast deposits of oil-soaked shales in Colorado. The question is how are we going to make that oil practically available.

[7] Head Engineer, Bureau of Ships, Navy Department, Washington, D. C. Mem. ASME.

It would seem, therefore, that the value of the paper would be enhanced by the presentation of one possible method which was found in the huge mass of "unfinished business" created by the German surrender. Apparently, only preliminary tests were completed, the purpose of the research having been the development of a "back-pressure" engine suitable for jet propulsion. Unless this idea is found to possess attractive economic possibilities on this side of the ocean, it will stay buried in the mountains

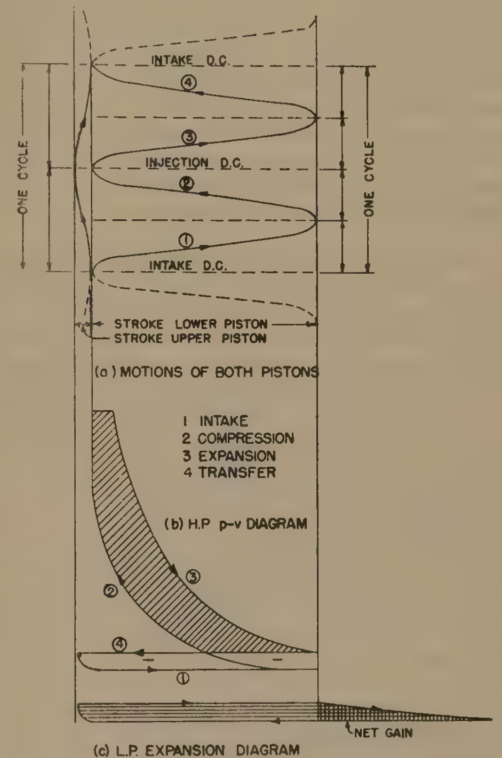


Fig. 6 Diagrams of German Back-Pressure Engine

of rubble and desolation by a people more concerned about physical survival than about contributions to the more efficient use of fuel resources of the world.

The concept of the new engine rests upon two opposing pistons, the lower connected to a crankshaft in a normal manner, the upper made of larger diameter and driven at one half the crankshaft speeds as from the customary camshaft of the four-cycle engine. The engine operates on a cycle requiring four strokes of the lower piston and two strokes of the upper piston. The accompanying diagram, Fig. 6, of piston motions illustrates most effectively the resultant operation. It is obvious that the pistons in the "intake" inner dead center must be separated by only the minimum possible physical clearance, and at the "injection" dead center by the required combustion space. Thus only cool and undiluted air will fill the cylinder at the end of the intake stroke, the cylinder volume in the lower dead center being in excess of piston displacement by one half of the combustion-space volume. It will then be compressed in the normal manner on the upstroke of the two pistons, the fuel will be injected at or near the dead center, followed by the expansion or power stroke down to the bottom dead center of the lower piston. As in every normal four-cycle engine, the exhaust valve is then opened and the gases of combustion are to be "transferred" by displacement. In a normal engine exhausting into the atmosphere, their energy content would be lost and wasted. Furthermore, the gases filling the clearance space normally remain in the cylinder and thus

not only is their heat energy wasted but they also dilute and reduce the incoming air charge by heating it. By "transfer" to the low-pressure turbine of a turbocharger, for instance, all of this energy can be recovered in the form of compressed charging air.

The new engine differs from others by this ability to displace all of the gases of combustion out of the cylinder. Furthermore, by properly designing the turbine nozzles and the exhaust duct between the cylinder and the turbine, the pressure of exhausting gases can be maintained and a very much greater part of the heat energy of these gases can be converted into mechanical energy of the turbine shaft, thereby raising the weight and the density or pressure of the air charge, with consequent increase in net output and improvement in thermal efficiency or fuel consumption.

This is one way of materializing the energy now going to waste in the exhaust gases of internal-combustion engines. Is it practically attractive? Or is there a better way?

E. F. Obert.[8] The authors have emphasized that either an ideal turbine or an ideal complete-expansion engine will deliver the same ideal amount of work when the working fluid is the exhaust gas from a reciprocating piston engine. The authors show that their first equation, for the complete expansion engine, gives the same answer as their fourth equation, for the ideal exhaust turbine. This proof (and the necessity for such a proof is debatable because the end result appears to be self-evident) is not too satisfying because their fourth equation must be solved by graphical integration. A more elegant solution is available by defining a slightly different thermodynamic system. Consider that their fourth equation arises from analysis of a nonsteady flow system, which consists of a turbine with flow from the cylinder into the turbine and from the turbine into the atmosphere. Applying the "first law" to each element of fluid which flows reversibly through the adiabatic turbine (which is the system)

$$\underset{s=C}{W_{\text{rev}}} = \int_{G_0}^{1} (h_x - h_0)dG = \int_{G_0}^{1} h_x dG - h_0(1 - G_0)$$

and this is the fourth equation of the paper. Suppose now that the system is defined as the "cylinder" at the time of exhaust blowdown. For this system the decrease in internal energy is

$$\Delta U = G_0 u_0 - u \dots\dots\dots\dots\dots\dots [1]$$

Now examine the fluid leaving this system, say, through a reversible nozzle

$$\underset{\text{rev}}{E_{\text{flow}}} = \underset{s=C}{\Sigma}\, G(u_0 + p_0 v_0 + \frac{1}{2g_c} V^2)$$

$$= \int_{G_0}^{1} h_0 dG + \int_{G_0}^{1} \frac{V^2}{2g_c} dG$$

$$= (1 - G_0)h_0 + KE_{\text{rev}} \dots\dots\dots\dots\dots\dots [2]$$

The change in energy indicated by Equation [1] herewith, for an adiabatic and reversible expansion must be equal to that of Equation [2] but of opposite sign

$$(1 - G_0)h_0 + KE_{\text{rev}} = u - G_0 u_0$$

Since the work to be obtained from this flow must come from conversion of the kinetic energy

$$W_{\text{rev}} = [u - G_0 u_0 - (1 - G_0)h_0]_{s=C} \dots\dots\dots [3]$$

Equation [3], unlike the author's fourth equation, is quite easily solved (and of course Equation [3] reduces directly to th author's first equation, because $G_0 = v/v_0$).

It is well to remark that the foregoing equations (and most o the author's equations) are not limited to perfect gases and len themselves to more exact solutions when using combustion chart such as the Hottel charts.

A. R. Rogowski.[9] The authors of this paper have introduce a valuable concept. The fact that an ideal blowdown turbin will convert into work the same amount of energy that the idea engine would, if both expansions were carried to ambient pres sure, gives the engineer a most useful criterion for comparison o actual exhaust-turbine performance.

Since the authors are considering ideal cases, and since, in fact the authors' expression for w_e is strictly true only if $v = v_1$, it i suggested that the exhaust release be assumed to take place a bottom center. Then the ideal efficiency of the blowdown proces will be

$$\frac{w_e}{(u - u_1)(1 - G_0)}$$

From a practical standpoint, early exhaust release will resul in loss of reciprocating-engine work which is chargeable agains the turbine. The values of T and p may be calculated easily fo the theoretical engine cycle, but are much more difficult to esti mate for an actual engine. This is another argument for usin the foregoing concept, mainly as a simple criterion for compariso of actual turbine performance.

A simple proof that the exhaust energy recovered by an idea blowdown turbine is equal to

$$w_e = (h - h_0) - A\,(p - p_0)v$$

may be obtained by considering the engine cylinder and turbin as an "open system."

Starting with the energy present per pound of gas in th cylinder at point (4) as u, then during the blowdown process th system will lose $(1 - G_0)$ pounds which leaves the system, carry ing with it $(1 - G_0)u_0$ Btu of internal energy. The flow work re quired to force it out of the system will be

$$Ap_0\,(1 - G_0)v_0$$

The system will lose turbine work w_t, and will gain piston worl during the exhaust stroke of

$$Ap_0\,(v_1 - v_2)$$

At the end of blowdown, there will remain G_0 lb in the cylinder the thermodynamic condition of which will be the same as th turbine exhaust, since all the gases have expanded isentropicall from p to p_0. The energy left in the cylinder will be G_0u_0.

Noting that $G_0 = v_2/v_0$ and $v = v_1$, we may write

$$u - w_t - (1 - G_0)u_0 - Ap_0\,(1 - G_0)v_0 + Ap_0(v_1 - G_0 v_0) = G_0 u$$

From which, combining terms

$$w_t = u + Apv - Apv + Ap_0 v_1 - (1 - G_0)h_0 - G_0 A p_0 v_0 - G_0 u_0$$

or

[8] Associate Professor, Mechanical Engineering, Northwestern Technological Institute, Evanston, Ill. Mem. ASME.

[9] Associate Professor of Aeronautical Engineering, Massachusett Institute of Technology, Cambridge, Mass.

$$w_t = h - Av\ (p - p_0) - (1 - G_0)h_0 - G_0h_0$$

Therefore

$$w_t = h - h_0 - A\ (p - p_0)v = w_e$$

QED.

A similar analysis may be made for the case of an internal-combustion engine discharging its exhaust gas into a receiver held continuously at p_4 pressure. From the receiver, the exhaust gases flow through the turbine, expanding to p_0 pressure. In this case

$$h_{rec} = h$$

Flow through the turbine $= (1 - G_0)$ lb per lb of gas in the engine cycle

$$\therefore \text{ turbine work per lb gas in cycle } = (1 - G_0)(h - h_0)$$

The extra pumping work required of the engine to force the gas into the receiver is

$$A\ (v - v_2)\ (p - p_1)$$

In this case

$$G_0 = \frac{v_2}{v} = \frac{1}{r} \text{ and } p_1 = p_0$$

$$\text{Pumping work } = A\ (v - G_0v)\ (p - p_0)$$

$$= A\ (1 - G_0)(p - p_0)\ v$$

The net turbine work per lb of gas in the engine cycle will thus be

$$(1 - G_0)\ [(h - h_0) - A\ (p - p_0)\ v]$$

or

$$\frac{r-1}{r}\ [(h - h_0) - A\ (p - p_0)\ v] = \frac{r-1}{r}\ w_e$$

Henry Schreck.[10] When presenting this paper, the authors seemed to limit their work to a blowdown process of a mass of gas out of a container through a nozzle down to atmosphere. The example used in the paper is for the exhaust of a gasoline engine, but reference is also made to the exhaust of Diesel engines which, therefore, are included in this treatise. Now then, in the case of a gasoline engine, say, an aircraft engine, there is an abundance of heat left in the outgoing exhaust, which is not true in the case of the Diesel engine, for which an actual example and its solution might be in place in the authors' closure.

A few years ago one of our leading locomotive builders entered into a contract with a well-known builder of supercharger exhaust turbines for airplane engines. The contract came about on the basis of an outstanding success of the latter in developing the same system for a new supercharged high-speed locomotive Diesel, using a liberal-sized exhaust manifold and the blowdown method as the authors call it. On the Diesel engine this system turned out to be a miserable failure because the turbine did not have sufficient output to meet the requirements for the desired supercharging. The condition was turned into success by substituting the larger exhaust manifold with small individual pipes from each cylinder, thereby utilizing the heat mass plus the kinetic energy of the gases as they leave the cylinders.

It would be interesting to have somebody, maybe the authors, work out the actual conditions with which the engineer is confronted when determining the exhaust-power utilization of an actual Diesel engine.

[10] Retired, Jackson Heights, New York. Mem. ASME.

H. W. Welsh.[11] In view of the current widespread interest in compound reciprocating engines, this paper is timely indeed. Theoretical studies of this nature are frequently quite valuable in analyzing the potentialities of various proposed designs. The chart presented in Fig. 3 of the paper is quite general since it can be applied to theoretical Diesel and Otto-cycle engines equally well.

However, the fact that Fig. 3 is based on the temperature and pressure at the time of release makes it extremely difficult to use for practical applications. Also, since the exhaust energy came from another cycle, the writer cannot help but wonder what is the relation between the work of the basic cycle and the convertible energy of the exhaust.

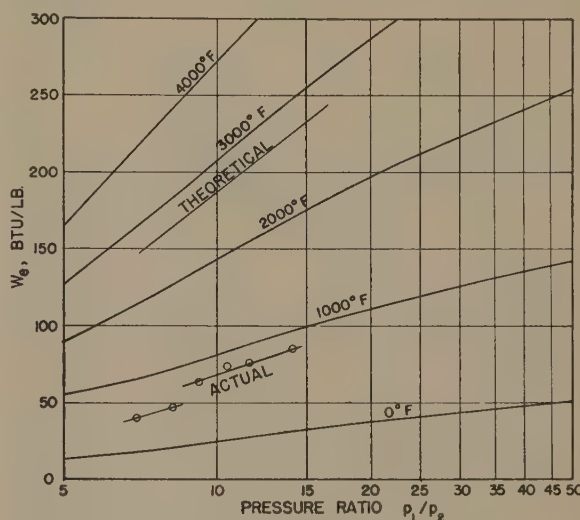


Fig. 7 Convertible Blowdown Energy C9HC Single-Cylinder Test Data

(Data taken at sea level; 2500 rpm; 0.080 F/A; 4.552 in.^{2} nozzle.)

Unfortunately, the authors have not presented test data showing the degree of correlation between theory and actual performance. For that reason, the writer takes the liberty of presenting some data obtained by his company. The tests were run to determine the actual convertible energy in the blowdown from cylinder to exhaust pressure.

The data, expressed as a function of theoretical cycle release to exhaust pressure ratio, are compared to the authors' calculations in Fig. 7 herewith. Although there is considerable agreement with regard to the slope of the curve, the recovery is only about 35 per cent of that predicted from Fig. 3. The large difference is indicative of the losses inherent in exhaust power-recovery systems. The losses cover almost every conceivable form, including shock, friction, and heat transfer.

Owing to the magnitude of the losses, it is apparent that actual test data must be used for all power-recovery applications and, while the chart in Fig. 3 is very interesting from the theoretical standpoint, it should not be considered a short cut to evaluating

[11] Assistant Project Engineer, Wright Aeronautical Corporation, Wood Ridge, N. J. Jun. ASME.

the energy actually available. The reader is referred to three papers which discuss the amount of available energy in the exhaust.[12]

In terms of engine performance, test data show that the blowdown energy (convertible energy in this paper) varies from 30 to more than 50 per cent of the brake output of typical aircraft engines. When this energy is converted to useful work in a turbine, geared to the engine, the engine power output can be increased to 15 to 25 per cent at no increase in fuel consumption. Although performance data are restricted, it may be of interest to note that the writer's company is now producing a true compound engine utilizing the blowdown energy. Pictures of the engine appeared in newspapers and trade magazines a few months ago.

Authors' Closure

The authors wish to express their sincere appreciation for the enthusiasm and co-operation of those who participated in the discussion, which is rather thought-provoking and adds much value to the paper.

Prof. Israel Katz described an interesting compound power plant for aircraft propulsion in which a two-stroke cycle engine is used as a "topping unit." The compounding of two-stroke cycle engines has been a relatively neglected subject in this country, but it does seem to present interesting possibilities.

Mr. E. C. Magdeburger described an intriguing engine which, it is understood, was to be used to create direct thrust on the water (jet effect) when the exhaust is discharged through a submerged nozzle. To this end the engine is to make use of three unusual features, i.e., (1) pushing out the residual gas from the clearance volume, (2) recovering blowdown energy, and (3) creating a back pressure in the exhaust at the expense of the piston work, utilizing that, too, to increase the thrust.

For an ordinary engine an elevated back pressure has a very detrimental effect upon its volumetric efficiency. If a "back-pressure engine," such as described by Mr. Magdeburger, were used, then the volumetric efficiency would be almost independent of the exhaust pressure. Such an engine, although requiring unusual valve arrangement, may be attractive for certain applications.

Profs. E. F. Obert and A. R. Rogowski offered proofs of the equation for convertible blowdown energy. The former's proof applies to two-stroke cycle engines, and the latter's to four-stroke cycle engines. The final result is the same in both cases.

The authors had obtained similar proofs but for brevity they were not included in the paper. However, the authors are glad that their omission has been ably filled by Professors Obert and Rogowski.

[12] "Design of Nozzles for the Individual Cylinder Exhaust Jet Propulsion System," by B. Pinkel, L. R. Turner, and F. Voss, NACA ACR April, 1941 (Wartime Report E-83).

"Performance of an Exhaust-Gas 'Blowdown' Turbine on a Nine Cylinder Radial Engine," by L. R. Turner and L. G. Desmon, NACA ACR E4K30 December, 1944 (Wartime Report E-30).

"Engine Compounding for Power and Efficiency," by E. F. Price and H. W. Welsh, SAE Transactions, vol. 2, April, 1948, pp. 316–328 and 344.

The authors agree with Professor Obert that most of the eq tions in the paper are not limited to perfect gases. They can used for solutions involving combustion products such as v done in obtaining Fig. 4.

Professor Rogowski introduced an expression for the "id efficiency of the blowdown process." His expression is not cl to the authors. The efficiency of a "process" is difficult to defi For practical purposes, Professor Rogowski's u and u_1 are at le as difficult to evaluate as the authors' T and p for an act engine.

Professor Rogowski presented an interesting scheme in wh an engine exhausts into a receiver where the pressure is k constant at p_4. This scheme is theoretically very attractive b also involves certain limitations. Primarily, it is applicable o to four-stroke cycle engines where one piston stroke is availa for pushing the exhaust gases out of the cylinder against p_4 pr sure. Then, owing to the necessity of maintaining the exha back pressure at p_4, which is much higher than the intake press p_1, any low or medium compression-rate engine is out of t question. Unless a high-compression-ratio engine or some spec engine, such as described by Mr. E. C. Magdeburger is used, t volumetric efficiency will be intolerably poor.

Assuming an ideal Diesel cycle with a compression ratio 20:1, a fuel/air ratio of 0.045, and initial conditions of 14.7 p and 560 R, the air-cycle efficiency is 54.6 per cent, and the th retical mean effective pressure is 188 psi. Based upon Profess Rogowski's scheme, the efficiency is raised to 61.7 per cent, a the mean effective pressure to 212 psi (referred to engine d placement). This shows a theoretical net gain of 13 per cent power and economy. This is paid for by the addition of a turbi and its controls.

In reply to Mr. Henry Schreck, it may be pointed out th Fig. 3 of the paper is equally applicable to gasoline engines a Diesel engines. A typical Diesel blowdown with 60 per cent cess air involves 50 psia release pressure and 1000 F release te perature, which results in a convertible blowdown energy of Btu. One pound of cylinder gas contained 19,000/23.2 820 Btu. The recovered energy is therefore approximately 4 p cent. Mr. Schreck also touched upon the question of exhau manifolding. Although the authors are carrying on a very pro ising experimental research on this subject, the test results a not yet released for publication.

Mr. H. W. Welsh's information is a welcome addition to t paper. While the authors agree with Mr. Welsh that actual te must be resorted to for exhaust-energy recovery applications, t fact that the actual recovery is only 35 per cent of that predict from Fig. 3 indicates that the art of designing exhaust-blowdo turbines is still in its infancy. As Mr. Welsh correctly poi out, the losses in a blowdown turbine cover almost every co ceivable form, such as shock, friction, and heat loss. A blowdo turbine also has to negotiate widely varying blade-to-jet spe ratios. It is hoped that concerted research will improve su stantially the efficiency of present-day exhaust-blowdown t bines.

Operating Experiences in Connection With Regenerative Reheat-Turbine Installations

By C. A. ROBERTSON,[1] MILWAUKEE, WIS.

This paper relates to the performance and design of regenerative reheat steam turbines, particularly several large 1800-rpm tandem-compound generating units. The author, who has been associated with field work and design of both these and marine types, discusses installations in the stations of two power companies. A description of these units, together with their respective reheat systems and protective equipment, is given, and many details of design and operation are discussed. The paper covers experiences for a period of 18 years, and also considerations relating to units now being installed or under construction.

Introduction

RENEWED interest in the reheat cycle for large steam turbines can be based not only on the inherent increase in thermal economy obtainable but also on the equivalent flexible performance between reheat and nonreheat units, as indicated by the 18-year operating experience of one large turbine manufacturer. Operating records of these regenerative reheat steam turbines have included base load and varying load operation, together with frequent start-up and shutdown performance. Protective equipment in the reheat lines has functioned reliably and the modifications which have been made relate more or less to details of design.

The earlier regenerative reheat units have been in operation for some 18 years. These are supplied by steam generated from a bank of boilers and a separate reheat boiler. Protective equipment in the reheat lines consists of intercepting and steam-unloading valves. These units have performed comparably with nonreheat units located in the same station.

An 80,000-kw unit of later design in another station, supplied from a single boiler with a resuperheating section, has set the highest standard of performance in the industry. Subsequent 80,000-kw reheat units in this station are in operation, being installed, or under construction.

Initial objections to the wide use of the reheat cycle for large turbines were based on a reluctance to increase the unit investment for reheat unless it enabled a reduction of exhaust moisture to below 12 per cent. At some higher steam pressures, reheat was the best answer to high exhaust blade erosion by moisture.

Today the reheat cycle enables an inherent gain in the thermal economy of large turbines which is variously estimated at from 4 to 6 per cent.[2] In view of greatly increased fuel costs, an individual analysis of investment in each specific case often completely justifies its use.

This paper discusses factually the important features of the operation of reheat turbines and should reduce further any remaining objections to application of the cycle.

Description of Units

The large steam turbines, which constitute the subject matter of this paper, are essentially similar in basic design and characterize the entire regenerative reheat-turbine design practice of this turbine manufacturer since installation of its initial reheat unit in 1930. This basic similarity lies in the use of a single-flow high-pressure reaction element and a low-pressure double-flow reaction element both coupled in tandem to an 1800-rpm generator. While recent construction involves impulse-reaction types, the four units in operation are reaction. The reheat belt in each case is in the middle of the high-pressure cylinder with a diaphragm between the outlet to and the inlet from the reheater.

Because of the presence of a large volume of steam in the reheat lines and reheater, protective devices for the reheat call for intercepting valves or the equivalent in the reheat lines. These intercepting valves are under control of the main speed governor and are arranged to close if the speed of the unit increases to approximately 3 or 4 per cent above normal. This arrangement has proved entirely satisfactory without causing undue complication of the governor gear.

Four turbines now in operation are discussed: 65,000-kw and 115,000-kw units in station A installed in 1930 and 1931, and two 80,000-kw units in station B installed in 1935 and 1943. Two additional units of 80,000-kw capacity which are under construction or being installed for station B are illustrated in section in Fig. 1. The earlier 80,000-kw units are illustrated in Fig. 2.

The additional units are designed for slightly higher steam conditions. The design of these is somewhat modified in that they are of the impulse-reaction type, and the low-pressure element is designed for $29^1/_2$ in. Hg vacuum.

Each station A unit is supplied from banks of boilers plus a reheat boiler. Steam conditions are, respectively, 650 psig, 750 F, reheat at 750 F; and 650 psig, 750 F, reheat at 750 F. Both operate condensing at 1 in. Hg exhaust pressure. Four stages of regenerative feedwater heaters are used for each turbine.

Station B units have single boilers with resuperheating sections, and steam conditions are, respectively, 1230 psig, 825 F with reheat at 825 F; and 1290 psig, 850 F with reheat at 850 F. Five stages of regenerative feedwater heaters serve each unit.

The design of all these reheat turbines was based on the consideration that the division of pressure at the exhaust of the high and the inlet to the low-pressure elements would be at approximately 20 psia, or nearly atmospheric pressure. The reheat stage was to be located in the high-pressure cylinder.

Construction of the high-pressure cylinder involves the use of a diaphragm separating the high- and low-temperature reheat steam and requires large extraction belts or passages, together with the fact that the high-temperature steam is located adjacent to the low-temperature steam. However, experience with this construction over a period of years has indicated the reliability of this type of design.

Large extraction belts for reheat steam and large reheat pipe connections have resulted in substantially no more difficulties

[1] Engineer in Charge, Test and Engineering Calculations Group, Steam Turbine Department, Allis-Chalmers Manufacturing Company. Mem. ASME.

[2] "High Fuel Costs Revive Reheating," by C. W. Bloedorn, *Power*, vol. 92, August, 1948, pp. 485–487.

Contributed by the Power Division and presented at the Annual Meeting, New York, N. Y., November 28–December 3, 1948, of The American Society of Mechanical Engineers.

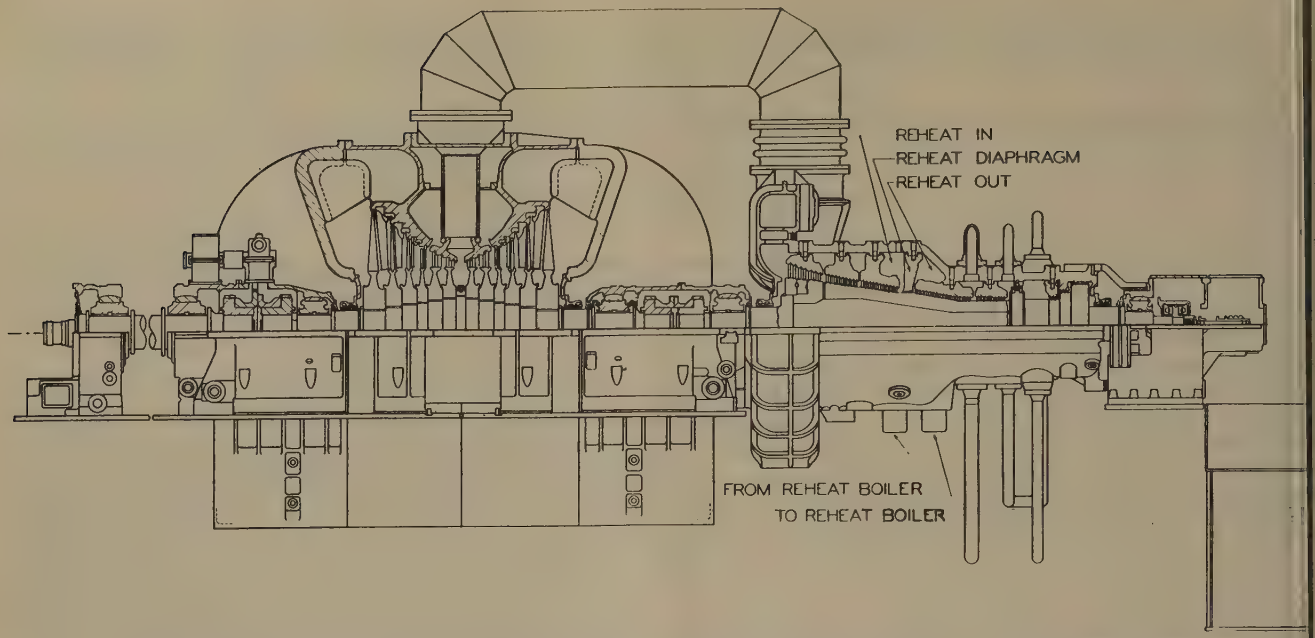


Fig. 1 Sectional View of the New 80,000-Kw, 1800-Rpm Regenerative Reheat Units in Station B

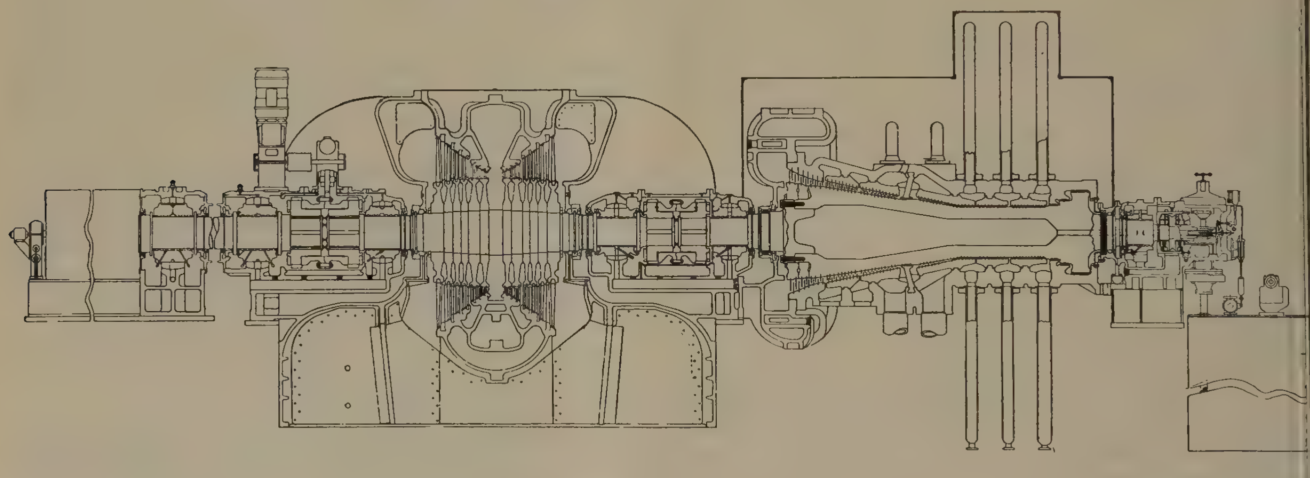


Fig. 2 Sectional View of the Two 80,000-Kw, 1800-Rpm Regenerative Reheat Units Now in Operation in Station B

than those offered by applying the regeneration extraction cycle to the turbines.

Differential expansions between the parts of the turbine rotor and cylinder resulting from the introduction of reheat within the cylinder have given no cause for concern, nor have they required any major design considerations not consistent with ordinary condensing-turbine design practice.

The magnitudes of these differential expansions as affecting the radial clearances were no more than in a straight condensing unit. The radial clearances of the blading were those for standard nonreheat design of similar size, and periodic inspections have indicated that these clearances have been maintained.

Axial differential movements of parts of the turbine rotors relative to the cylinder, especially those in the exhaust end, or reheat section, never gave the designer serious difficulties. The larger movements occurred only during starting periods, and these were provided for in the original design.

This type of construction of having the reheat belt in the center of the high-pressure cylinder has several advantages, as follows:

1 It maintains the conventional straight condensing form c cylinder design characterized by unidirectional flow, and th maintenance of small diameter in the high-pressure high-tempera ture region and gradually increasing diameter toward the low pressure end of the turbine.

2 The leakage losses through the diaphragm are maintaine relatively low because the pressure drop across the diaphragm i small, amounting to the pressure loss of the reheater and con necting pipe and valves.

3 With approximately atmospheric pressure at either end c the casing, the shaft-packing problem is comparatively simple.

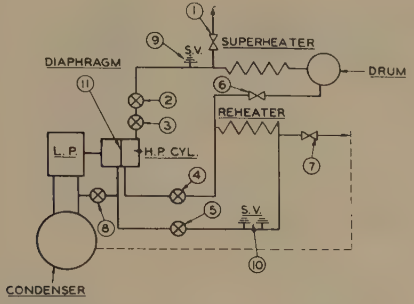
Design of Controls

For ordinary condensing turbines, speed-governor control o main inlet valves, and an overspeed governor operating to clos the throttle valve, together with check valves on extraction line to feedwater heaters, are sufficient to prevent excessive overspeed ing of turbine in case of sudden loss of load.

With reheat units, however, the large volume of steam store

in the reheat piping and steam space of the reheater has an enormous potential energy which is not controlled by the main inlet valves. Therefore it is necessary to provide some form of reheat protective equipment, such as intercepting valves and unloading valves, in the reheat lines; and for these steam turbines these valves are under control of the speed governor.

The intercepting valves, Fig. 5, and also Fig. 3, items (4) and



1 Superheater drain valve to condenser
2 Throttle valve
3 Governor-operated control valves
4 Reheat intercepting valve, oil-operated by governor
5 Reheat intercepting valve, oil-operated by governor
6 Reheater by-pass valve manually operated
7 Reheater drain valve to condenser
8 Steam-unloading valve between reheat and condenser
9 Superheater safety valves
10 Reheater safety valves
11 Reheat diaphragm within high-pressure cylinder

Fig. 3 Schematic Diagram of Unit Boiler per Turbine Installation in Station B

(5), are wide open at normal speed, and begin to close at a predetermined overspeed of about $1^1/_2$ cycles above normal frequency, or 1850 rpm.

The closing time of these valves is faster than that indicated on the tests of earlier units (see Fig. 17), designed for lower steam conditions, which required greater valve travel.

The unloading tests reported in this paper show that, in order to limit the overspeed rise, the essential idea is to start these valves closing very soon after the tripping off of the load and not wait until the speed rises to actuate the overspeed governor or mechanism.

As mentioned, the reheat protective valves are set to operate at $1^1/_2$ cycles above normal frequency. This setting, based on experience, appears to be quite satisfactory as most line-frequency variations are well below this value.

Should the line frequency for a very brief interval increase to this value, or slightly higher, there may be a slight movement of the valves; however, as the interval is usually short, the valves are restored to normal setting.

On the more recent installations, the intercepting valves have a definite partially closed position for overspeeds between 3 and 5 per cent above normal frequency.

Should the frequency rise as a result of line disturbance to, say, $2^1/_2$ cycles above normal, the valves would only close to a fixed position and remain there instead of closing entirely. After the frequency drops, the valves again open.

In addition to the intercepting valves, a steam-unloading valve, Fig. 6, is also provided, which under normal load operation is closed and steamtight. This valve, however, following a load trip-out, opens by the action of the speed governor slightly after the intercepting valves begin to close. This valve relieves the steam in the reheat line at a point between the intercepting valve and the turbine, and by-passes it to the condenser, see Fig. 3, item (8).

The purpose of this valve is to reduce the stage pressure at the reheat inlet and also to keep this pressure near condenser value so that very little steam flows through the turbine blading between reheat and low-pressure exhaust, thereby reducing to an absolute minimum the possibility of turbine-speed acceleration due to leakage steam or residual steam.

On the units in station A, the reheat control consists of a single intercepting valve with two steam-unloading valves, Fig. 15, items (6) and (7), respectively. All three are oil-operated and are under governor control. They function at a predetermined overspeed of approximately 1850 rpm. This arrangement has proved quite successful, based upon unloading tests, and under actual emergency conditions.

In the later station B, the units which are at present operating are equipped with two intercepting valves, Fig. 3, one in each reheat line to and from the boiler, together with an unloading valve, thereby isolating the turbine from the storage volume of steam in the reheat system on the boiler side of the intercepting valves.

These are oil-operated and are under governor control. They function at a predetermined overspeed of approximately 1850 rpm. Emergency trip-out has demonstrated satisfactory performance.

Protective Equipment for New Units for Station B

For the new units under construction for station B, Fig. 3 illustrates schematically the arrangement of the reheat cycle, including the boiler and turbine. The arrangement of the governing system, including the reheat control, is shown in Fig. 4.

The general idea involved in this protective equipment is the same as that on the two units now operating. The operation of the intercepting valves is the same in this respect, that the valves are spring-loaded for closing and that they are opened and held by relay oil pressure. The essential differences are in the matter of details. The arrangement also includes a steam-unloading valve tapped into the return line from the reheater between the intercepting valve and the turbine.

Figs. 5 and 6 are included for the purpose of illustrating the type of valves for intercepting and unloading, respectively.

In this plant, with the unit arrangement of one boiler per turbine which is so well suited to the use of reheat, Fig. 3 shows schematically the arrangement of the reheat cycle, including boiler and turbine.

In the design of the protective equipment for a unit arrangement of one boiler per turbine, there are a few general ideas which are worth mentioning:

1 Fundamentally, the protective equipment in a plant, with a unit arrangement of one boiler per turbine, should be interrelated with respect to both boiler and turbine.

2 Turbine overspeed, which may be almost as destructive as a boiler explosion, seems properly the principal concern when designing and preparing operating instructions for a reheat unit.

3 Protecting the turbine transcends by far the need for protecting the reheater and superheater upon loss of load.

4 Following load trip-out, obviously the fuel feed in the boiler must be stopped very promptly.

5 Referring to the unloading tests which are described in this paper, leakage of steam into the turbine during overspeeding following load trip-out, is undesirable from the standpoint of holding the maximum overspeed within safe limits.

6 Modern large steam-turbine units, equipped with hydrogen-generator-cooling, have a no-load resistance that is extremely low; therefore relatively small quantities of steam in the neighborhood of 10,000 lb per hr, or 1 or 2 per cent of full-load flow can cause undesirable overspeeding.

7 The reheater and its lines can be drained to the condenser, thus obviating the need of throttling intercepting valve No. 5 to remove condenser vacuum from the reheater system. Besides

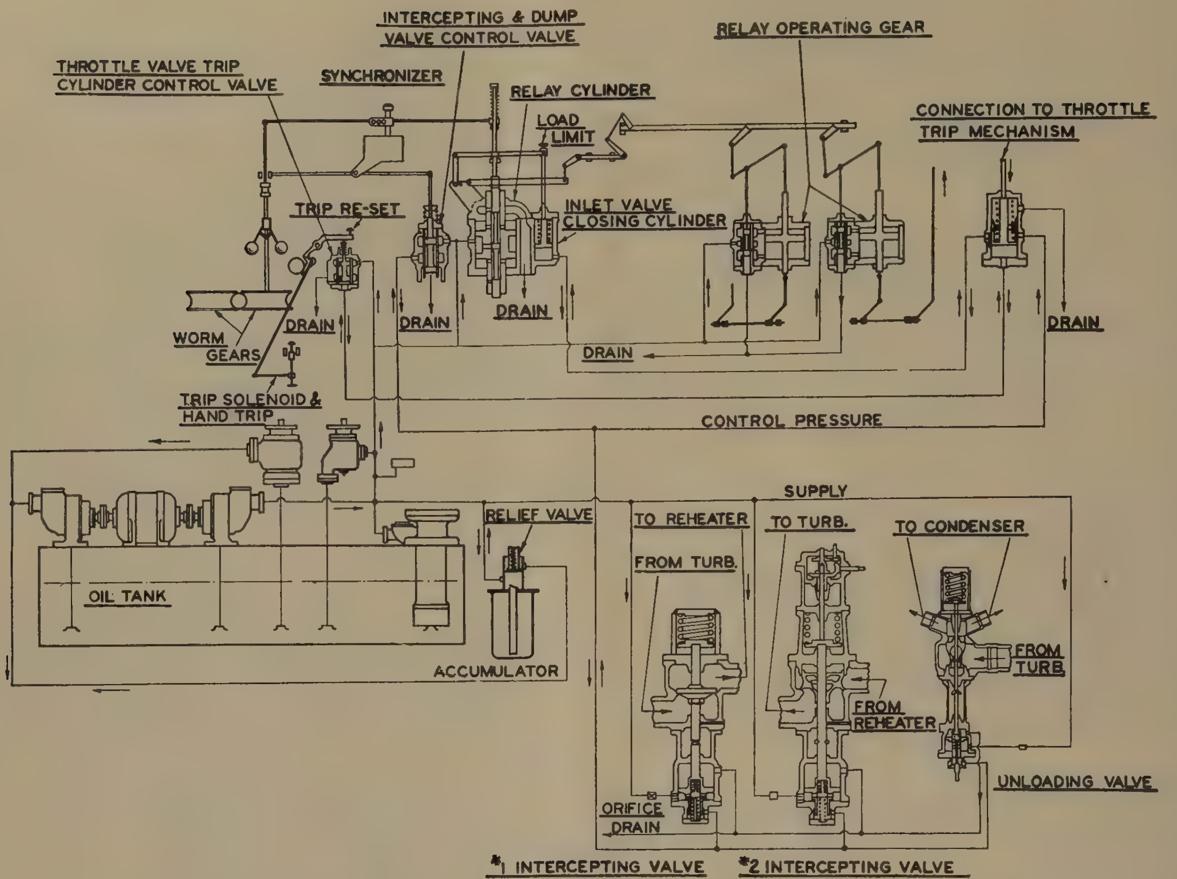


Fig. 4 Arrangement of Governing System Including Reheat Control in Station B

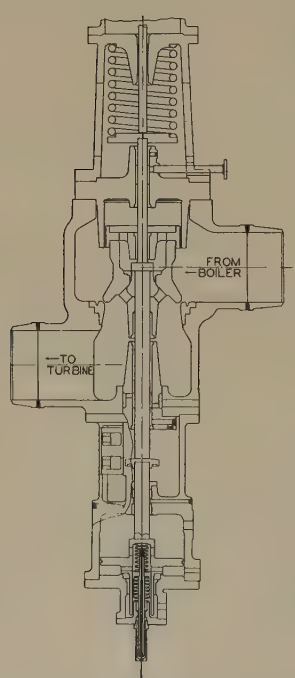


Fig. 5 Oil-Controlled Intercepting Valve

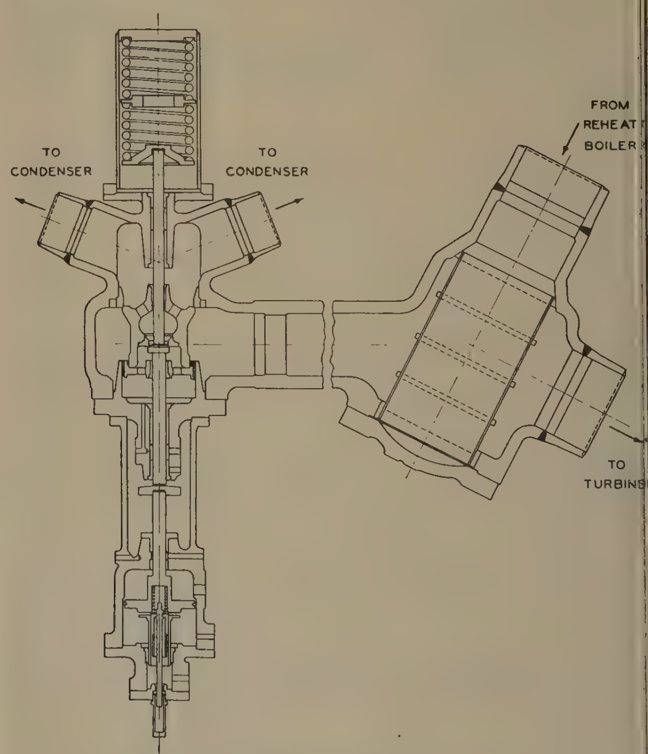


Fig. 6 Oil-Controlled Unloading Valve

drains to the condenser never cause turbine overspeeding. If the reheater and its lines are not drained to the condenser, but drained to the atmosphere, then, in order to prevent air getting into the reheat lines during starting periods, intercepting valve No. 5 must be throttled to hold the pressure in the lines and reheater to a value higher than atmospheric, say, 5 to 10 psig. The intercepting valve, item (4), Fig. 3, is designed to act as a relief valve in case the pressure in the high-pressure cylinder should increase too greatly. For instance, at any time this intercepting valve should be in a closed position, and the inlet valves or throttle valves should allow steam to pass into the cylinder, the steam pressure would not rise to boiler pressure, but valve No. 4 would act as a relief valve and relieve the steam pressure within the cylinder to the reheat system.

Interrelationship of Reheater and Turbine in Plant With Unit Arrangement of One Boiler per Turbine

Referring to Fig. 3, a schematic arrangement of a reheat cycle including boiler and turbine, some designs provide a means of protecting the reheater upon loss of load during accidental tripout of the unit by a manually operated reheater by-pass valve, and often an emergency pressure-reducing valve at the location of item (6), which may be set to open when the flow to the high-pressure turbine is stopped by either the throttle or governing inlet valves (2) or (3) being closed.

Consider the case where an emergency pressure-reducing valve is installed in parallel to valve (6), and upon loss of load this emergency valve opens following the closing of valves (2) and (3).

The intercepting valves (4) and (5) will close, bottling up the steam in the reheater and connected piping. The result of flow, therefore, through the emergency pressure-reducing valve located at (6) would build up the reheat pressure to a point where the reheater safety valves would open.

This would aggravate the condition should any leakage occur through the intercepting valves (4) and (5) with regard to turbine overspeeding, because the reheater pressure would be higher. Lowering of the reheat pressure would be more desirable.

As an alternative solution, drain valve (7), connected to the reheater, could possibly be profitably opened to help guard against leakage of either intercepting valves (4) and (5).

It seems that the proper location for an automatic emergency reducing valve would be in parallel with drain valve (7). The steam in the reheater and reheater piping would then be relieved to the condenser, producing cooling circulation in the reheater.

Consider the arrangement in this station. The volume of steam in the reheat piping between intercepting valves and reheater amounts to almost 4 times the volume in the reheater. One half of this volume is in the reheater piping between intercepting valve (4) and reheater. This relatively cool steam in this pipe is the quantity that would produce circulation and cooling of the reheater should valve (7) be opened.

Hand operation of valve (6) would be less dangerous than automatic operation. However, the turbine operator only should direct its opening and only when the turbine speed is held well below normal.

In so far as the superheater element is concerned, upon loss of load, steam flow through superheater will be provided by superheater safety valves.

Based upon the experiences in this station it can be stated that, obviously, fuel feed must be stopped very promptly upon loss of load. Instantaneously with its discontinuance, there need be no concern of superheater or reheater tubes overheating. These relatively heavy tubes have entirely adequate heat capacity for the few seconds elapsing between shutting of the turbine valves and stopping of fuel feed.

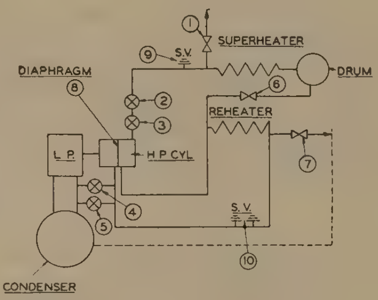
Sometimes it seems apparent that too much attention has been devoted to superheater and reheater cooling and too little to turbine-overspeeding safety.

There seems a real basis for manufacturers and users both insuring that never are any high-pressure lines so connected that their discharge steam can pass through any turbine blading during emergency conditions. The turbine throttle and inlet valves upon closing must not be by-passed in any conceivable manner.

Unit Boiler per Turbine Installation With Special Type of Reheat Control

A special arrangement of reheat protective valves, shown schematically in Fig. 7, has been suggested to the author.

The difference between this type of protective equipment and that shown in Fig. 3 is the elimination of the intercepting valves from the reheat lines, and the substitution of very large steam-unloading valves connected between the reheat line and the condenser, as indicated in Fig. 7, items (4) and (5). The control of these valves would be substantially the same as that described and indicated in Figs. 3 and 4. These valves would begin to open at about 2 and 3 per cent overspeed and would be wide open between 4 and 5 per cent overspeed.



1 Superheater drain valve to condenser
2 Throttle valve
3 Governor-operated control valves
4 Steam-unloading valve between reheat and condenser
5 Steam-unloading valve between reheat and condenser
6 Reheater by-pass valve manually operated
7 Reheater drain valve to condenser
8 Reheat diaphragm within high-pressure cylinder
9 Superheater safety valves
10 Reheater safety valves

Fig. 7 Schematic Diagram of Unit Boiler per Turbine Installation With Special Type of Reheat Control

The effectiveness of this type of reheat protective equipment would be in the speed at which the steam in the reheater system could be discharged into the condenser, in relationship to the no-load resistance offered by the unit under consideration, or commonly referred to as the WR^2, or mechanical inertia of the rotating elements.

Regenerative reheat steam-turbine units of the 1800-rpm type, designed for high steam pressures and high vacuum, which means high mechanical inertia in the rotative parts, together with relatively small volume in the reheater system due to the high density of the steam, would be better adapted to this type of control than would steam-turbine units designed for higher speed and smaller mechanical inertia.

Applying this arrangement of large-capacity steam-unloading valves as the sole protective equipment in the reheat system, to 1800-rpm, 80,000-kw, tandem-compound, regenerative reheat steam-turbine units, the resulting maximum overspeed upon tripping or loss of load would appear to be well within the desired safe limits.

These unloading valves would be under control of the main speed governor and would relieve the stored reheat-system steam directly to the condenser.

In station B the volume of the reheat system is about 1250 cu ft. At full load, the reheat pressure is about 385 psia and corresponds to about 750 lb of steam.

Following the trip-out and during the period when the steam-unloading valves are open, the division of steam through the turbine blading, after the reheat diaphragm, is approximately 17 to 20 per cent, the difference being by-passed through the unloading valves to the condenser.

The rate of steam flow to the condenser for a brief interval of time may be as high as 3 times normal full-load steam flow, resulting in a drop in vacuum of the order of 6 or 7 in.

The 1800-rpm type of steam-turbine generator units discussed in this paper have the higher order of mechanical inertia.

The stored energy in the reheat system is the same for both 1800 and 3600-rpm units of equal capacity.

With the same kinetic energy in either case, and the protective equipment requiring the same interval of time for operation, the resulting overspeed for the 3600-rpm type would be somewhat of the order of twice as high in percentage as for the 1800-rpm type.

In other words, the 3600-rpm type of unit would require the protective equipment to function in about one half the time interval of that for the 1800-rpm type of units in order to maintain the same percentage of overspeed.

As an added protective feature, it might well be mentioned that the units in both stations A and B have an electrical tripping mechanism which trips the throttle valve and the reheat protective equipment immediately upon the opening of the main generator circuit breaker in emergencies. In this type of trip, the reheat protective equipment operates immediately and does not wait for a speed rise to initiate its operation by means of the speed governor.

Limitations of Protective Equipment Consisting of Intercepting Valves

In the following discussion the conventional type of reheat protection by means of two intercepting valves and one steam-unloading valve is compared with a proposed arrangement of protective equipment requiring only two unloading valves.

1 Intercepting valves do not become effective in the closing direction until the valves are almost closed.

2 Intercepting valves introduce pressure drops in the reheat line, thus decreasing the total available energy and increasing the heat consumption of the unit.

3 Fouling of the intercepting valve (for instance, foreign metal or particles underneath the seat) would cause the valve to be open partially and leak, thus creating a condition for excessive overspeed upon loss of load.

4 Where intercepting valves are used, reheat safety valves are necessary in the reheat lines to protect the reheat system following the period of emergency unloading when the intercepting valves are closed.

Relative Advantages of a System Using Only Large Unloading Valves

1 Steam-unloading valves become effective as soon as they start to lift off the seats.

2 Unloading valves do not introduce a pressure drop in the reheat system.

3 Fouling of the unloading valves (for instance, foreign metal or particles underneath the seat) would cause the valve to be partially open and leak. However, this would not create a condition that would result in excessive overspeeding upon loss of load. Of course, however, a valve of this type, if it leaks, would require immediate maintenance to eliminate loss of economy.

4 Where unloading valves are used and no intercepting valv these unloading valves would open following a loss of load or tri ping; therefore the present battery of reheat safety valv item (10), Fig. 7, together with all attendant expense, could dispensed with.

5 Where two steam-unloading valves are used as shown, iter (4) and (5), Fig. 7, either one of which alone will prevent ove speeding, they are inherently more reliable than two intercepti valves and one steam-unloading valve. Like dual throttle valve failure of either of the two intercepting valves creates a hazard.

6 An objection to installation of steam-unloading valves di charging to the condenser, would be the tripping off of the hou auxiliary supply at the same time as the main unit trip-out. I this case the circulating-water pumps would stop, resulting po sibly in sudden loss of vacuum.

Trends in Regenerative Reheat Steam Turbines

The list of regenerative reheat steam turbines given in Table presents 18 years of reheat steam-turbine experience of one tu bine manufacturer.

TABLE 1 REGENERATIVE REHEAT STEAM TURBINES IN STALLED BY AUTHOR'S COMPANY, INDICATING TRENDS I STEAM CONDITIONS

Rating, kw	Station	Unit no.	Steam conditions	Date installed
65000	A	4	650 psig, 750 F, 1 in. Hg abs; reheat to 750 F	1930
115000	A	5	650 psig, 750 F, 1 in. Hg abs; reheat to 750 F	1931
80000	B	1	1230 psig, 825 F, 1 in. Hg abs; reheat to 825 F	1935
80000	B	2	1290 psig, 850 F, 0.5 in. Hg abs; reheat to 850 F	1943
80000	B	3	1290 psig, 850 F, 0.5 in. Hg abs; reheat to 850 F	1948
80000	B	4	1380 psig, 900 F, 0.5 in. Hg abs; reheat to 900 F	Under con struction
8000 shp	Marine propulsion regenerative reheat steam-turbine unit		1200 psig, 740 F, $1^1/_2$ in. Hg abs; reheat to 740 F	1942

The increase in steam conditions, as shown in chronological or der, illustrates the trend in the industry to obtain better heat per formance in view of rising fuel costs.

Higher thermal efficiencies were also desired in the marine in dustry, and working with the author's company, a pioneer high pressure regenerative reheat unit of 8000 shp was installed on a cargo ship.

The unit was of three-cylinder construction with reheat be tween the high-pressure and the intermediate-pressure turbines a a pressure of 230 psig and 740 F temperature at full load.

The high-pressure element, of the impulse type, was rated a 8012 rpm; the intermediate-pressure element, of the impulse-reaction type, was rated 5000 rpm; and the all-reaction low-pressure element was rated at 4200 rpm.

Starting Procedure for Regenerative Reheat Steam-Turbine Units

Over a period of many years the author's company has investigated and developed certain procedures in starting and shutting down a number of its larger steam-turbine units, both reheat and nonreheat.

The methods developed cover the starting of steam-turbine units for two extremes of conditions, namely, starting from cold, or starting following a short shutdown.

As the industry is installing more and more large units, the frequent shutting down and starting of these units will be necessary. This will also apply to reheat units; therefore the question as to the comparative flexibility of reheat to nonreheat units becomes an important factor.

In view of this situation, the author is including in this paper a brief summary consisting of graphs and explanations relating to starting of reheat units and a tabulation comparing this performance with that of nonreheat units.

The data presented strongly indicate that the flexibility of the reheat units with reference to frequent starting and shutting down is fully suitable for modern central-station service.

The graphs portray a procedure common to both types of units.

Table 2 indicates the comparison of normal starting time required for various sizes of units ranging from 25,000 to 80,000 kw, three of which are nonreheat units and two are regenerative reheat units.

TABLE 2 STARTING TIME FOR SHORT SHUTDOWNS

RATING	TOTAL STARTING TIME	TURNING GEAR	SPEED	DESIGN CONDITIONS
25 000 KW.	30 MINUTES	YES	1800 RPM	350 PSIG-675F.-29IN.VAC.
35000 KW.	30 MINUTES	YES	1800 RPM	350PSIG-675F.-29IN.VAC.
50000 KW.	30 MINUTES	YES	1800 RPM.	600PSIG-725F.-29 IN.VAC.
65000 KW.	30 MINUTES	YES	1800 RPM.	600 PSIG - 725 F. 725 F. REHEAT-29 IN.VAC
80000 KW.	30 MINUTES	YES	1800 RPM.	1230 PSIG-850 F. 850 F.-REHEAT-29IN.VAC.

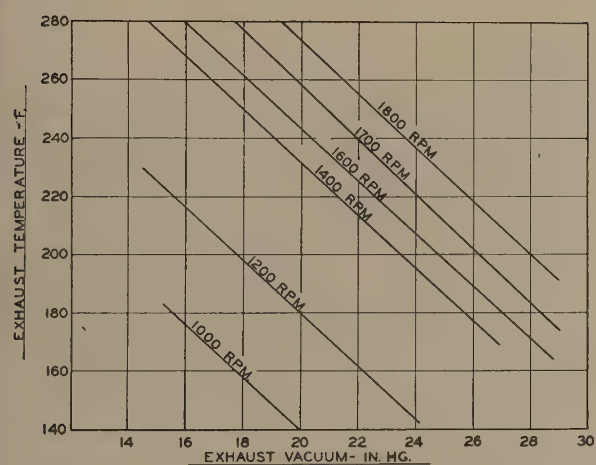


FIG. 8 EXHAUST-STEAM TEMPERATURES VERSUS EXHAUST VACUUM FOR DIFFERENT SPEEDS, 80,000 KW REHEAT UNIT
(1230 psig, 850 F, 850 F reheat, 29 in. Hg vacuum.)

With regard to the regenerative reheat units, one was of the single boiler per turbine type; the other was served by a number of boilers interconnected.

Fig. 8 is a graph showing the relationship between exhaust temperature and vacuum for different speeds during the starting period. This graph indicates the necessity of obtaining high vacuum during the latter part of the starting period in order to prevent excessive exhaust temperatures.

Fig. 9 shows two different starts for an 80,000-kw reheat unit, illustrating the application of the data shown in Fig. 8.

These two starts were made on succeeding days. In the first, the exhaust temperature got out of control, necessitating the shutting down of the unit to allow the exhaust to cool. The reason for the excessive exhaust temperature was caused by the inability to obtain the required vacuum at the higher speeds.

In the second start, the speed was increased more slowly, allowing the vacuum to increase to a higher value, thus reducing windage losses at the exhaust blading and thereby preventing an excessive exhaust temperature.

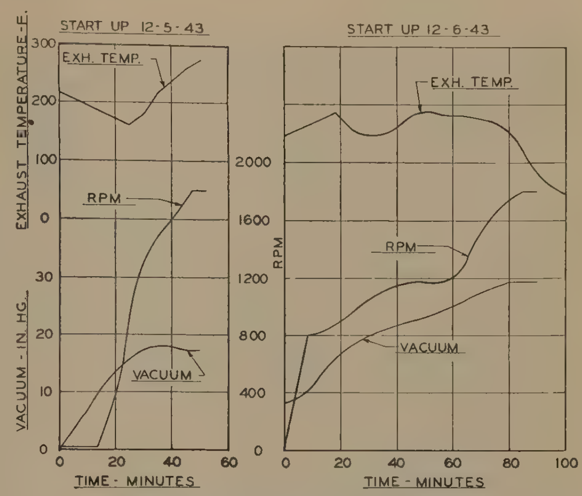


FIG. 9 STARTING PERFORMANCE, 80,000-KW REHEAT UNIT
(1230 psig, 850 F, 850 F reheat, 29 in. Hg vacuum.)

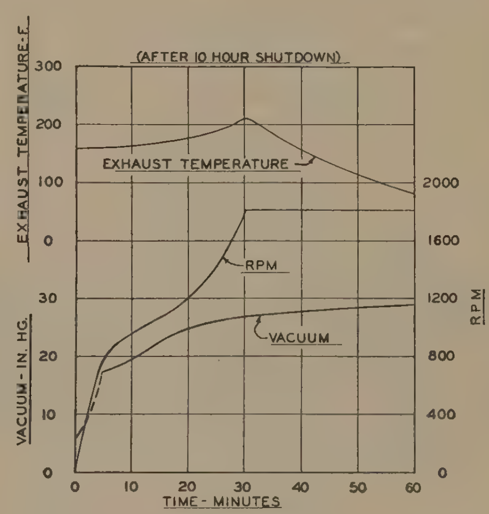


FIG. 10 NORMAL START-UP FOR SHORT SHUTDOWNS, 80,000-KW REHEAT UNIT
(1230 psig, 850 F, 850 F reheat, 29 in. Hg vacuum.)

The blade-tip speed on this unit is only 930 fps. Hence it can be seen that, with increasing blade speeds, the necessity of obtaining the required vacuum is of prime importance.

Fig. 10 shows the normal start after a short shutdown for the unit in Fig. 9. It indicates that practically full vacuum is obtained by the time full speed is reached. The maximum exhaust temperature is slightly above 200 F.

Fig. 11 shows the start of the same unit after a long shutdown, for instance, a start following a scheduled inspection.

The unit operates on a unit system with one boiler, and the reheat section integral with the boiler. In starting, the boiler and the turbine are brought up together.

This turbine is equipped with water-seal glands, and it will be noted from the graph that the speed is increased initially very rapidly up to about 800 rpm, which is the sealing speed of the water-seal glands. This is done so as to reduce air leakage and increase the vacuum very fast.

From the graph the time scale starts from 120 min. The first 3

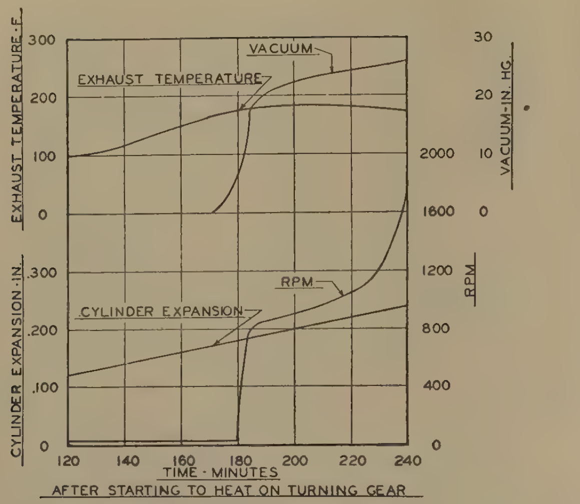


Fig. 11 Normal Starting Procedure From Cold, 80,000-Kw Reheat Unit
(1230 psig, 850 F, 850 F reheat, 29 in. Hg vacuum.)

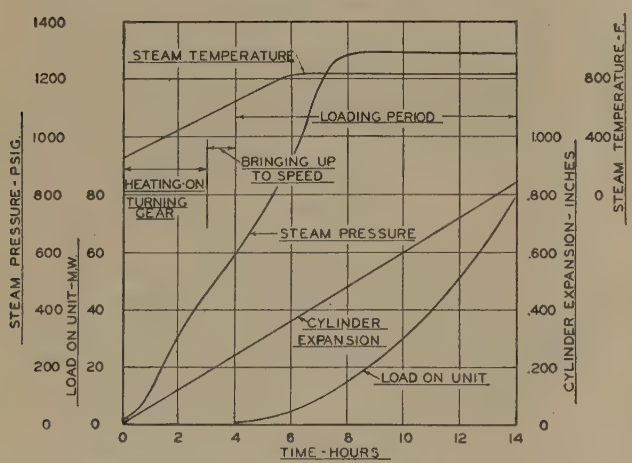


Fig. 12 Normal Starting Procedure From Cold, 80,000-Kw Reheat Unit
(1230 psig, 850 F, 850 F reheat, 29 in. Hg vacuum.)

hours are consumed in heating the turbine with throttle steam the unit is on the turning gear.

Thermal stresses introduced because of the great temperature difference between steam and metal are highest when the unit is while cold and hot steam is admitted to the turbine cylinder.

To control the thermal stresses within desired limits, a means is provided for measuring the rate of expansion of the turbine cylinder at the high-pressure end.

Fig. 12 gives a more complete starting record of the unit covered in Fig. 11, from the time the steam is admitted until the unit is fully loaded.

From this chart the time for heating on the turning gear and bringing up to speed is indicated.

The steam pressure and temperature are brought up uniformly during the starting period and the early part of loading.

During the cold starts, the essential factor to watch is that of keeping the thermal stresses of metal of the turbine within desired limits. This can be accomplished by observing the expansion of the turbine cylinder at the high-pressure end, and holding the rate of expansion to a fixed value per hour, for instance, 0.060 in. per hr normal, or 0.120 in. in emergencies. The rate of loading therefore becomes a function of expansion as indicated in th graph.

The total expansion is indicated by a straight line, and th loading is indicated by a curved line. About 10 hr are consume in increasing the load from zero to full load.

The time required for cold starts is not a critical factor, a these occur probably only once in a year or once in a longer perio

The adherence to this idea of maintaining minimum therma stresses to established limits reduces maintenance of turbines.

Table 3 gives the starting method, depending upon the time th unit has been shut down, but it is based on the initial expansion o the cylinder at the starting time relative to that of a cold unit From the table it will be noted that with 0.100 in. initial expan sion of cylinder, the starting time is 2 hr and 45 min, and wit 0.500 in., the initial expansion, the starting time is 30 min.

TABLE 3 RECOMMENDED STARTING TIME RELATIVE T INITIAL CYLINDER EXPANSION OF 80,000-KW REHEAT UNIT

INITIAL CYLINDER EXPANSION-INCHES	TIME BRINGING UP TO SPEED	TIME HEATING ON TURNING GEAR	TOTAL STARTING TIME
.100	45 MINUTES	2 HOURS	2 HOURS 45 MIN.
.200	40 MINUTES	1 HOUR	1 HOUR 40 MIN.
.300	35 MINUTES	$\frac{1}{2}$ HOUR	65 MINUTES
.400	30 MINUTES	$\frac{1}{4}$ HOUR	45 MINUTES
.500	30 MINUTES	0	30 MINUTES

[a] 1230 psig, 850 F, 850 F reheat, 29 in. Hg vacuum.

Normal Start-Up of a Unit Boiler Reheat Plant

1 Drain superheaters, reheaters, and piping adequately, without unnecessary loss of steam.

2 Start fire in boiler.

3 If lines are cold, heat up high-pressure lines and also warm up reheat lines by closing intercepting valves and by-passing high-pressure steam into one of the two reheat lines to allow this steam to circulate through reheat lines, the drains being connected to the condenser and the excess steam valved to a low-pressure header. Heating may be performed with only a low pressure in the boiler.

4 If the turbine is cold, the warming-up can be started as soon as the boiler pressure reaches a few pounds, and by means of this early start, time can be saved accordingly. The initial heating normally for a cold turbine is about 3 hr; however, the time is somewhat governed by the expansion of the turbine cylinder to a value of approximately 0.150 in.

5 Following this initial heating period, in case of a cold start, the turbine is started rolling and the speed is increased in accordance with the method described in the first part of this section under starting.

6 After the turbine has been brought up to speed, synchronize on the line and apply load on the unit as shown in Fig. 12, if the start-up is from cold. In case of a short shutdown, the rate of loading can be applied much faster, as the metal temperatures of the turbine casings and rotor are relatively high, and the change in thermal expansions is of smaller magnitude.

7 For normal overnight or week-end shutdown, the procedure is the same as for a nonreheat unit, the load being gradually reduced to zero, the unit tripped out and put on the turning gear after coming to rest. The steam pressure in the boiler is not deliberately reduced in any case, because resumption of service on start-up promptly encourages retention of boiler pressure, in-

stead of its useless dissipation. Steam lines are adequately above any saturation temperatures to require draining. Economy is improved by making all possible attempts to preserve or hold boiler steam pressure.

Functioning of Reheat Protective Equipment

Earlier in this paper the author described the reheat-control equipment, consisting of governor-operated intercepting and unloading valves in the reheat system, which operate in case of a condition where the electrical load is suddenly tripped off.

About 16 years ago the power company at station A performed a series of unloading tests on both of its reheat units. Figs. 13 and 14 show sectional views of the steam turbines. An elaborate test setup was made to determine the operation and sequence of the governor, pilot valves, intercepting valves, unloading valves, and the speed of the turbine during the unloading tests.

These tests were made at fractional as well as at full load, and this established the relationship between light-load and full-load conditions. Since that time, the practice has been to make light-load unloading tests following inspections, and the results of these tests indicate whether the protective equipment is in satisfactory working order.

Fig. 15 illustrates schematically the arrangement of the reheat cycle, including boilers and turbines. It applies to both reheat units in this plant.

Fig. 16 shows the method of operation of the control equipment. Items (7) and (8) are the spring-loaded intercepting and unloading valves, respectively, in the reheat system. The intercepting valve (item 7) is kept normally open, and the unloading valve (item 8) is kept normally closed by relay oil pressure.

In case of overspeed, the main speed governor (item 9) rises to a predetermined height and opens the oil control valve (item 15). When the oil control valve (item 15) opens, it releases the oil pressure underneath the oil pistons of the intercepting valve (item 7) and unloading valve (item 8), allowing them to move under force of their springs.

In case the speed increases sufficiently to operate the emergency trip, the overspeed governor (item 10), in addition to closing the throttle valve, operates on the 4-way oil trip valve (item 11), which in turn releases the oil pressure from underneath the oil pistons of valves (items 7 and 8), thus causing the intercepting valve to close and the unloading valves to open. The main inlet valves also will be closed by action of the overspeed governor operation at tripping speed, provided these oil-operated inlet valves are not already closed by direct action of the main speed governor.

As the turbine speed returns to normal, the intercepting and unloading valves, being under control of the speed governor, return to their normal operating positions, and the unit is in a position to be resynchronized the same as a conventional nonreheat unit.

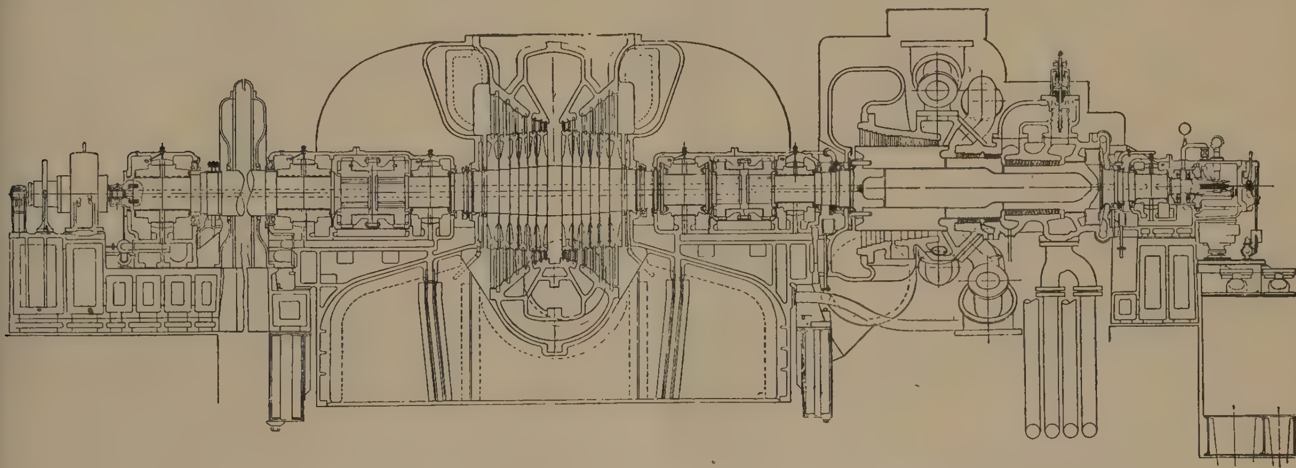


Fig. 13 65,000-Kw, 1800-Rpm Regenerative Reheat Unit in Station A

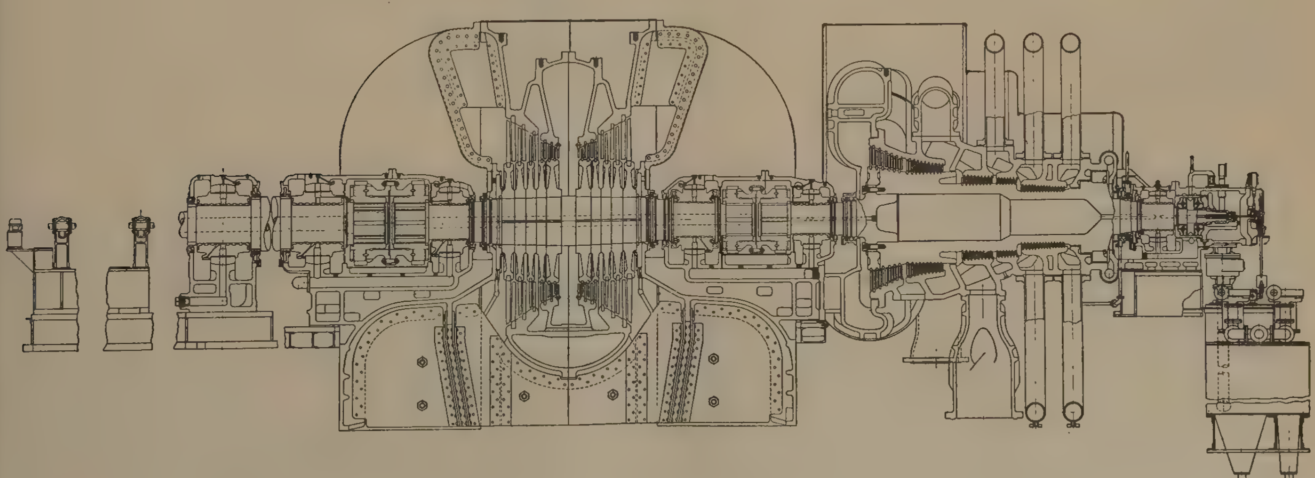
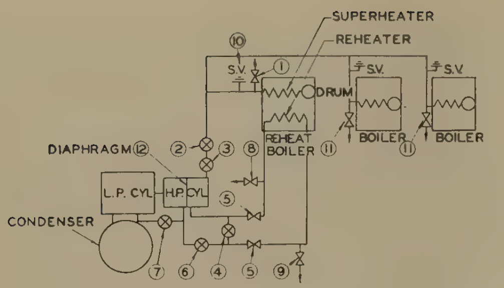


Fig. 14 115,000-Kw, 1800-Rpm Regenerative Reheat Unit in Station A



1 Superheater drain valve
2 Throttle valve
3 Governor-operated control valves
4 Reheat crossover valve
5 Reheat boiler shutoff valves
6 Reheat intercepting valve oil-operated by governor
7 Steam-unloading valve between reheat and condenser
8 Atmospheric relief valve or safety valve
9 Reheater drain valve to condenser
10 Superheater safety valves
11 Superheater drain valves
12 Reheat diaphragm within high-pressure cylinder

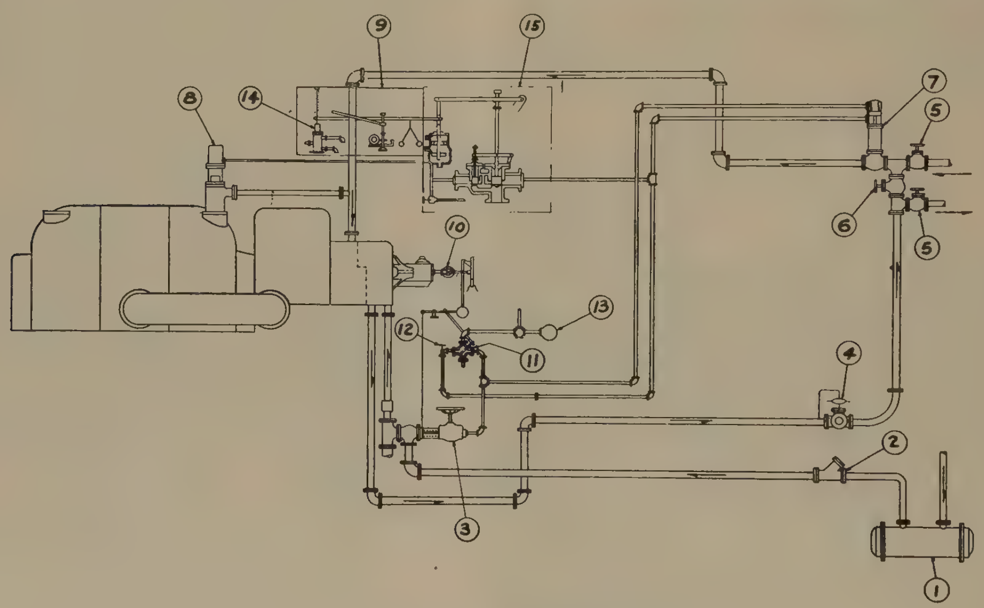
Fig. 15 Schematic Diagram of Interconnected Boiler, Reheat Boiler, and Turbine in Station A

close early and the rate of closing of these valves must be as fast a possible.

In these tests it will be noted that the steam-unloading valve Fig. 15, item (7), were set to open at an early moment of this se quence of operation.

Note also that the unloading valves were open for a brief inter val at the same time that the intercepting valve was open. There fore reheat steam at normal pressure passed through these tw valves directly to the condenser without going through the tu bine. The time interval was brief; however, the rate of discharg of reheat steam to the condenser was very high. The effect on th condenser vacuum was scarcely noticeable.

In Fig. 18 two curves are shown illustrating graphically th speed-limiting effect caused by the addition of a steam-unloadin valve, Fig. 15, item (7). The upper curve represents the charac teristic speed rise when the reheat protecting equipment, in thi case, consisted of only one intercepting valve, Fig. 15, item (6) and the unloading valve, Fig. 15, item (7), was omitted. Th lower curve represents the characteristic speed rise when the re heat protective equipment, consisting of both intercepting valve item (6), and the unloading valve, item (7), Fig. 15, was in service



1 Separator
2 Strainer
3 Throttle valve
4 Pressure-actuated relief valve
5 Stop valve to reheat boiler
6 Crossover valve; reheat line
7 Intercepting valve
8 Governor-controlled steam-unloading valve between reheat and condenser
9 Main speed governor
10 Overspeed stop
11 Four-way oil-trip valve
12 Needle oil-suppy valve
13 Oil accumulator
14 Oil-operating governor gear
15 Speed governor-operated oil-control valve for intercepting valve and steam-unloading valves

Fig. 16 Arrangement of the Governing System Including Reheat Control in Station A

Good operating practice would of course call for an investigation into the cause of the trip-out.

Unloading Test Results

The graphic characteristic curves of one of the unloading tests made at full load on one of the reheat units in station A is shown in Fig. 17. The purpose of inserting this graph is to show the sequence of operation of the valves, rise in turbine speed, and drop in reheat steam pressure, all plotted as a function of time.

Note the rapid rise in speed following load trip. Therefore, to decrease this rate of speed rise, the inlet and intercepting valves, Fig. 15, items (3) and (6), respectively, must begin to

The difference in speed rise amounted to about 130 rpm by the introduction of an unloading valve installed as shown in Fig. 15.

The closing times, Fig. 18, of the inlet valves and the intercepting valves were very nearly in agreement with each other; however, the speed continued to rise after the valves were closed. The reason for this is due to leakage steam passing through part of the turbine. Since there was only one intercepting valve isolating the return of reheat steam, the steam pressure on the high-pressure side of the diaphragm is maintained at reheat pressure in the reheat boiler; therefore the sources of leakage which tend to increase the speed after valves are, all closed are, namely: (*a*) leakage through the reheat diaphragm; (*b*) leakage through the laby-

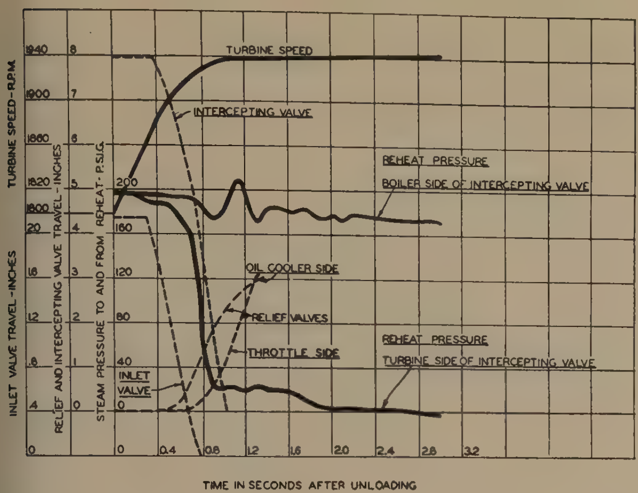


FIG. 17 UNLOADING TEST AT FULL LOAD ON 65,000-KW UNIT IN STATION A

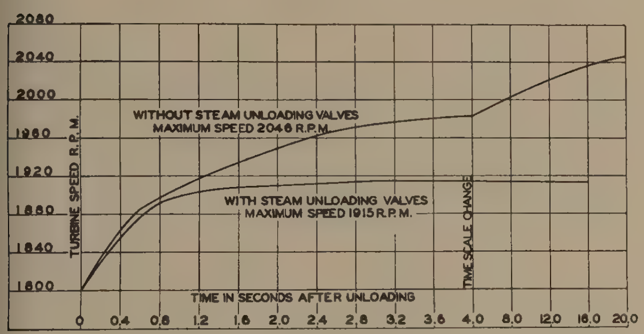


FIG. 18 COMPARISON OF SPEED RISE WITH AND WITHOUT UNLOADING VALVES

rinth steam packing in the balance pistons of the high-pressure unit; (*c*) possible leakage through the reheat intercepting valve.

With the addition of a steam reheat unloading valve, item (7), Fig. 15, connected between the reheat pipe and the condenser, the stage pressure on the low-pressure side of the diaphragm was maintained close to condenser pressure, thus reducing the available energy of leakage steam that passes through the blading into the condenser.

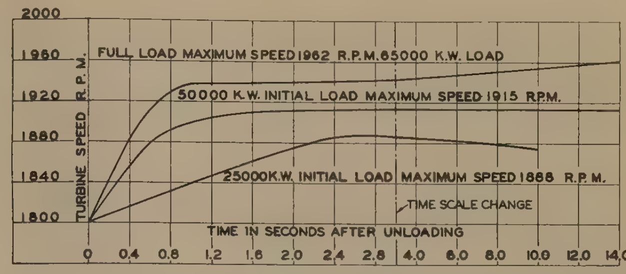


FIG. 19 COMPARISON OF SPEED-RISE CURVES DURING UNLOADING TESTS FOR DIFFERENT FRACTIONAL LOADS

Note that the unloading-valve connection to the reheat pipe is between the intercepting valve and the turbine on the return reheat line from the boiler; thus the steam passing through this valve, item (7), Fig. 15, is relatively small when other valves, items (3) and (6), Fig. 15, are closed.

Where no steam-unloading valve is used, two intercepting valves, one in each reheat line, see items (4) and (5) Fig. 3, are frequently used. In this way the volume of reheat steam trapped in the reheat lines cannot get back into the turbine to cause overspeeding except by leakage through the intercepting valves themselves.

Fig. 19 shows graphically the relationship of the rate of speed increase following unloading for different loads, ranging from 40 per cent fractional load to full load.

ACKNOWLEDGMENTS

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Developments in Resuperheating in Steam Power Plants

By E. E. HARRIS[1] and A. O. WHITE,[2] SCHENECTADY, N. Y.

Both steam and gas reheaters have been used for a number of years, and this paper is devoted to a discussion and evaluation of the gains in thermal efficiency as reflected in turbine heat rate realizable in modern steam turbines for central stations. The gains due to reheat are justified economically with present equipment and fuel costs. They represent an additional gain in fuel economy over and above the normal annual gain.

Introduction

RESUPERHEATING, or as it is more commonly called, reheating has been applied to steam power plants of all types and sizes for a great many years. If steam is expanded in successive steps from the initial to the final exhaust conditions as in a multiple-expansion reciprocating engine or in a turbine, the steam may be taken away from the engine (or turbine) at one or more points, reheated by high-pressure steam, or in a direct-fired reheater (including reheaters built into the main steam generator or boiler), and readmitted to the engine with a resulting increase in thermal efficiency, as evidenced by a reduction in the heat consumption per unit of output.

While both steam reheaters and direct-fired (or as they are sometimes called, gas reheaters) have been used, this paper considers primarily the case of the gas reheater. While steam reheating gives a reduction in heat rate, the gain is entirely nonthermodynamic, that is, the gain is due to a reduction in moisture losses in the engine and not due to improvement in the cycle efficiency as such.

This paper will attempt to evaluate the gains which can be realized by reheating under various conditions, and enable an evaluation to be made of the optimum reheater pressure, temperature, reheater pressure drop, and the effect of the efficiency of the various sections, as applied to modern steam turbines used in central-station and large industrial power plants, with a typical feedwater-heating arrangement.

Basic Considerations

The basic cycle is the well-known Rankine cycle, in which the working fluid is expanded adiabatically (i.e., constant entropy), from the initial temperature and pressure to the final or exhaust pressure, with a resulting decrease in total heat and the performance of external work. Heat is then given up at constant temperature, condensing the working fluid and rejecting heat from the cycle. The fluid (as a liquid) then has heat added, first along the saturated liquid line at variable temperature, then at constant pressure and temperature to vaporize the working fluid, and then generally further heat is added at constant pressure to superheat the working fluid. This cycle is shown diagrammatically in Fig. 1, and it can be shown that the work done is proportional to area *ABCDEA* and the thermal efficiency is area *ABCDEA* divided by

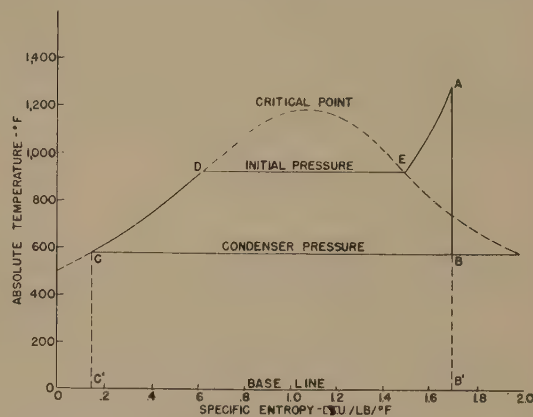

Fig. 1 Temperature-Entropy Diagram of Simple Steam-Plant Cycle

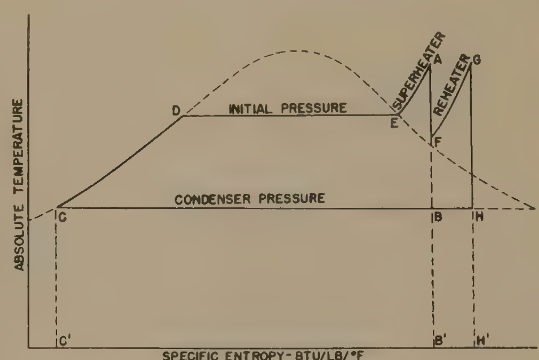

Fig. 2 Temperature-Entropy Diagram of Reheat Cycle

area *AB'C'CDEA*. In everyday usage the thermal efficiency of an ideal cycle is

$$\frac{3412.75}{(H - h) \times \text{TSR}}$$

where *H* is initial enthalpy, *h* is condensate enthalpy, and TSR (theoretical steam rate) = 3412.75/AE, AE being the available energy, which for steam may be read from the ASME Theoretical Steam Rate Tables. This is of course without extraction for feed heating. The corresponding efficiency when heating the feedwater in an infinite number of heaters, and with 100 per cent engine efficiency, may be derived from Selvey and Knowlton's paper (1).[3]

If now the working fluid be extracted from the engine at some point, say, *F*, Fig. 2, and heat added to resuperheat it to some point *G*, and then the expansion continued to *H*, the exhaust

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[3] Numbers in parentheses refer to the Bibliography at the end of the paper.

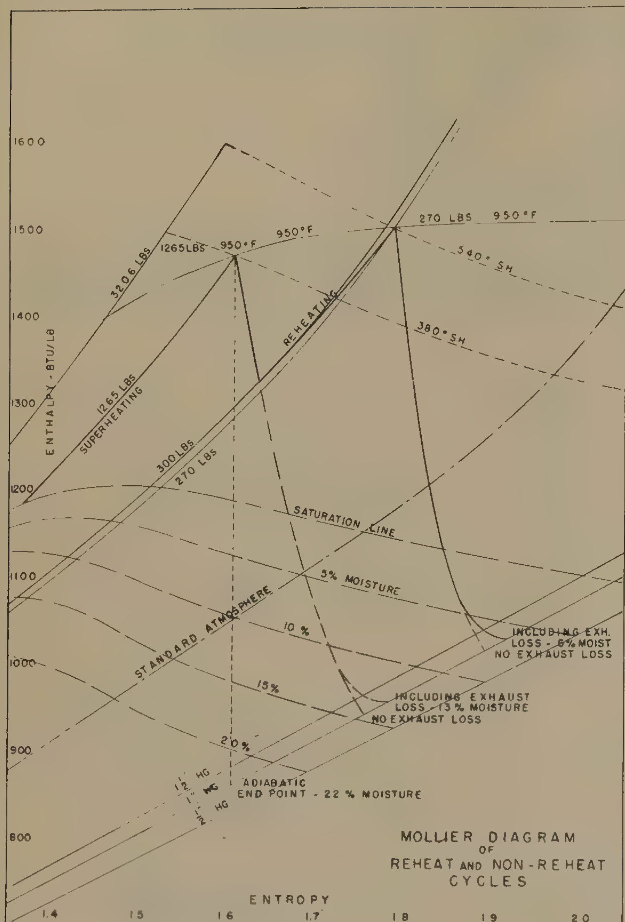

Fig. 3

pressure, a reheat cycle is defined. The work done in the cycle is now proportional to area *AFGHCDEA*, and the thermal efficiency is again *AFGHCDEA* divided by area *AFGH'C'CDEA*. If the diagrams are drawn to scale the increase in the ratio of the areas can be seen. The efficiency in the same terms as the foregoing, and on a nonextraction basis is

$$\frac{3412.75}{\dfrac{3412.75\,(H_1 - h) + 3412.75\,(H_3 - H_2)}{\text{AE}}}$$

where

H_1 = initial enthalpy at point A
H_2 = enthalpy at point F
H_3 = enthalpy at point G
h = liquid enthalpy at point C
AE = sum of adiabatic energies from point A to F and G to H.

For the ideal cycle, with reheat, an extension of Selvey and Knowlton's method can be used to obtain the theoretical heat rates with regenerative feed heating in an infinite number of heaters.

Theoretical heat rates have been calculated for various steam conditions and various reheater pressures and temperatures, and the gains over the regenerative Rankine cycle are plotted in Figs. 4, 5, 6, and 7. This gain is then the pure thermodynamic gain. The actual gain is greater than this by reason of the improvement in engine efficiency due to reduction in moisture loss, etc., and it is this total gain that is evaluated in the balance of the pape

Before showing the total gains to be expected, it might be well devote a few words to the nonthermodynamic gains. First, a probably the most important is the reduction in moisture los Several authorities (2, 3), have given the general rule that in steam turbine there is a loss in used energy (work done per pour of steam) in the moisture region of 1.15 per cent for each 1 per ce of average moisture during the expansion. An inspection of t Mollier diagram in Fig. 3 shows that for the reheat cycle, not on is the average moisture content during the expansion less, but le of the work is done in the moisture region, so that there is marked increase in the used energy for a given available energy c the turbine, which is, for convenience, taken account of as an i crease in the turbine efficiency from the reheat point to the e haust. Fig. 23 gives the correction factor to be applied to turbi efficiency for various initial superheats and pressures (see A pendix).

In addition to this gain, the increase in available energy r quires less steam flow for the same output. While this means tha effectively the reheat machine is slightly smaller than a nonrehea unit, and so should have a slightly poorer internal efficiency, the is also for the same last-stage annulus area, a smaller volume flo to the condenser, with a resulting lower leaving velocity and co responding exhaust loss. This gain can be taken advantage of i either of two ways: (*a*) use the same last-stage combination c the reheat as would be used on the nonreheat unit and obtain a increase in efficiency due to the reduction in exhaust loss; (reduce the area of the last-stage combination, which will result i a slightly smaller and less expensive turbine. The authors hav chosen alternative (*a*) to use in the calculations for this paper sinc at least in their company, last-stage combinations vary in rela tively large size steps, and so in actual practice the same last-stag combination would be used on either a reheat or a nonreheat ma chine of given rating.

Supplementary Advantages

The reduction in steam flow previously mentioned also permi a reduction in the size of the boiler, condenser, feedwater heater and related piping, and it is hoped that papers by representative of station-equipment manufacturers will discuss this phase of th subject. While it may not always be possible to take advantage c these gains, due to building column spacings, etc., they are, neve theless real and can, in general, be credited to the reheat cycle an help offset the increased cost of the reheat turbine and boiler.

One additional advantage, which in some cases might be signifi cant, is that the reduced heat rejection requires less cooling-wate supply for the same rating, or, where the cooling-water suppl is strictly limited, will enable the installation of from $7^1/_2$ to 1 per cent more capacity in reheat than in nonreheat units.

Basis of Comparison

The gain in heat rate due to reheat has been evaluated by ac tual calculation of heat rates for a nonreheat and a reheat cycl with a heater arrangement as shown in Fig. 8. Turbine efficienci have been kept as comparable as possible, and are based upon th curves in Warren and Knowlton's paper (4) with a uniform ex haust loss of $4^1/_2$ per cent of the shaft output for all nonrehea units at 1 in. Hg abs exhaust pressure, and an equal annulus are for reheat units at the same initial conditions. This assump tion, which results in a nonthermodynamic gain, is part of th gain due to reheating and is in the order of 1.2 per cent.

The assumptions for the heater cycle are indicated in Fig. 8 and the resulting gain in heat rate for the four initial pressures an the various temperatures considered are plotted versus reheate pressure in Figs. 9, 10, 11, and 12. These curves are all at 1 in

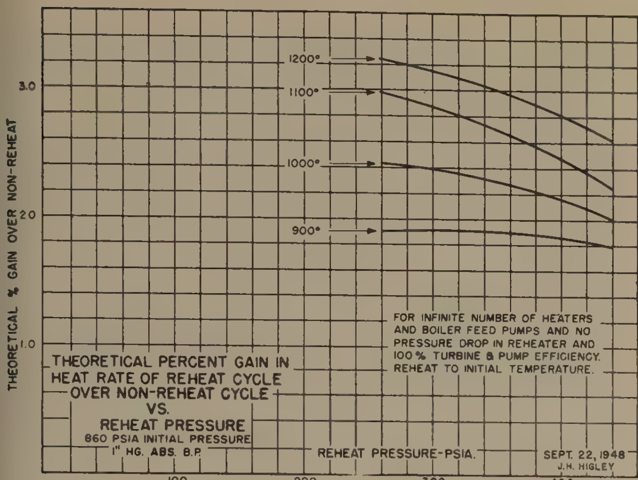

Fig. 4

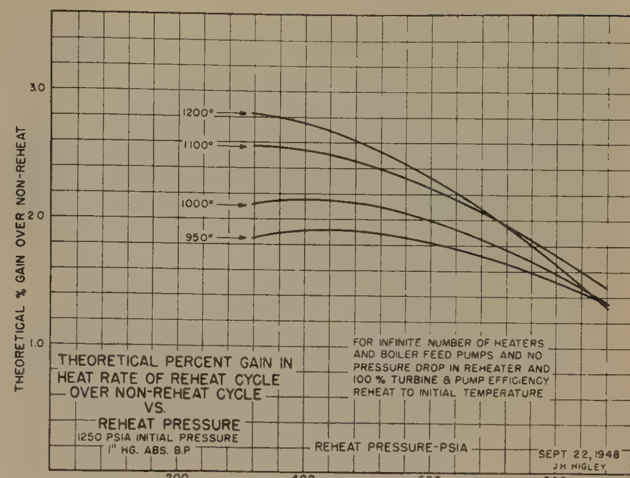

Fig. 5

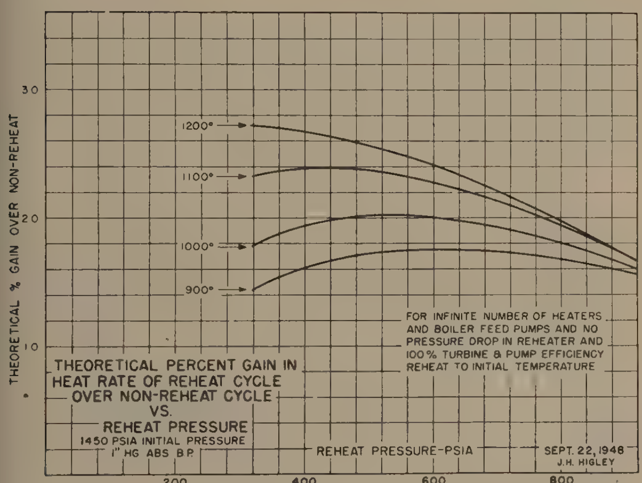

Fig. 6

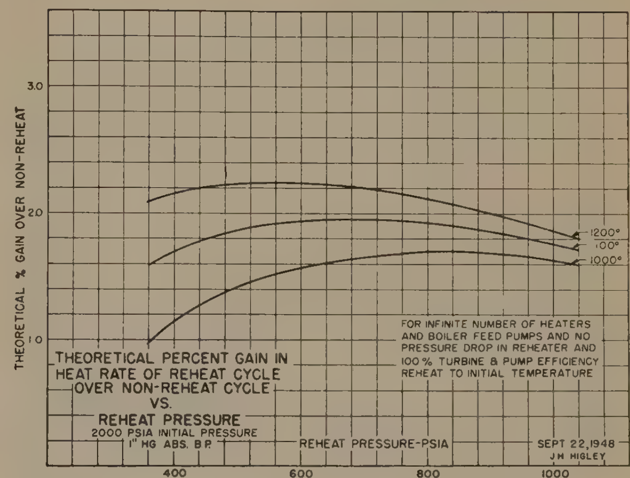

Fig. 7

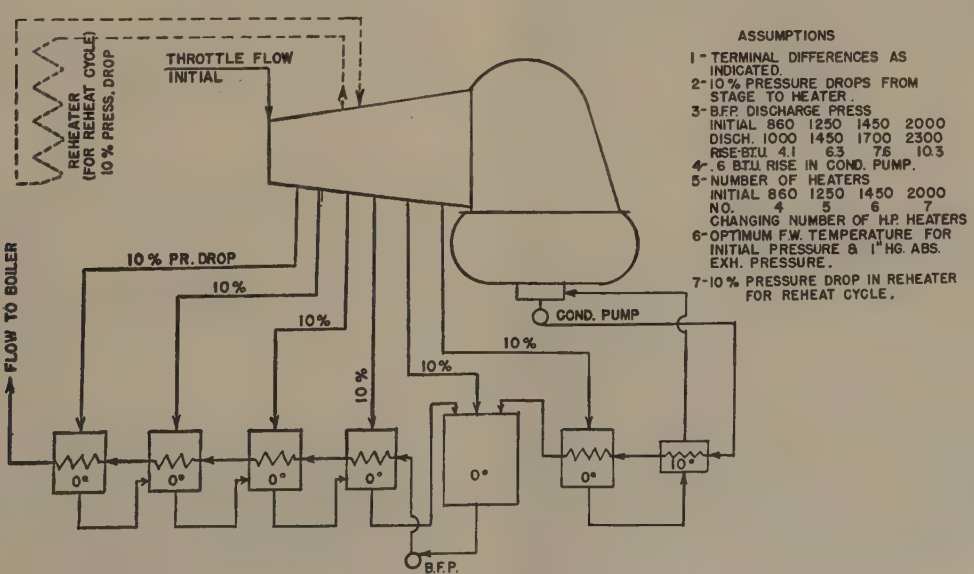

Fig. 8 Heater Arrangement

Hg abs exhaust pressure and optimum feedwater temperature as indicated by Salisbury (6) for each initial pressure, with reheating to the initial temperature in a reheat boiler. The gain due to the reduction in boiler feed-pump work with reduced flow in the reheat units is included. One of the heaters may be either above or below the reheat point. As this heater shifts from above to below the reheat point a discontinuity of the gain curve will be obtained. This shifting is indicated by the break in the per cent gain curves. The heat rates are defined as follows:

For nonreheat, Btu per kwhr =

$$\frac{F\ (H - h_f)}{(\text{Generator output}) - (\text{input to boiler feed pump})}$$

For reheat, Btu per kwhr =

$$\frac{F(H - h_f) - F_R(H_R - H_e)}{(\text{Generator output}) - (\text{input to boiler feed pump})}$$

where

F = throttle flow, lb per hr
F_R = reheater flow, lb per hr
H = initial enthalpy
H_R = enthalpy after reheater
H_e = enthalpy to reheater
h_f = actual enthalpy of feedwater entering boiler

Resultant Gains and Their Variations

An inspection of the curves, Figs. 9, 10, 11, and 12, of gain due to reheat reveals that on the average a reduction of 6 to 7 per cent i heat consumption over a nonreheat plant is obtained by rehea ing to the initial temperature, providing that the reheat pressu is near the optimum for modern steam-turbine plants.

The optimum reheat pressure is a function of the initial stea conditions, particularly the pressure. The curves of gain due t reheat are nearly coincident for various initial and reheat tempera tures, and the reheat pressure at which the maximum gain occu varies from 0.10 to 0.18 of the initial pressure.

The actual gain at the optimum reheat pressure varies fro 5.15 per cent with 860 psia, 900 F, and 1 in. Hg abs back pressure, t 6.9 per cent at 2000 psia, 1200 F, and 1 in. Hg abs back pressure The gain is a function of steam conditions, turbine efficiencies, pe cent exhaust loss, feed-heating arrangement, and reheater pres sure drop. The gains shown by the curves are realized in rehea units where the efficiency of the high-pressure section is equal t that assumed in the calculations, and the feedwater is heated t the optimum temperature.

Since the heat rate with reheat is dependent upon the efficienc of the various sections of the turbine, calculations of the effect o the efficiency of the high-pressure section of the turbine (down t the reheat point) were made. These show that at usual rehea pressures there is a reduction in heat rate of about 0.16 per cen for each 1 per cent increase in the turbine efficiency. However this relation will vary from 0.1 to 0.4 per cent for each 1 per cen change in high-pressure turbine efficiency, depending upon th reheater pressure. Fig. 13 shows per cent change in heat rat

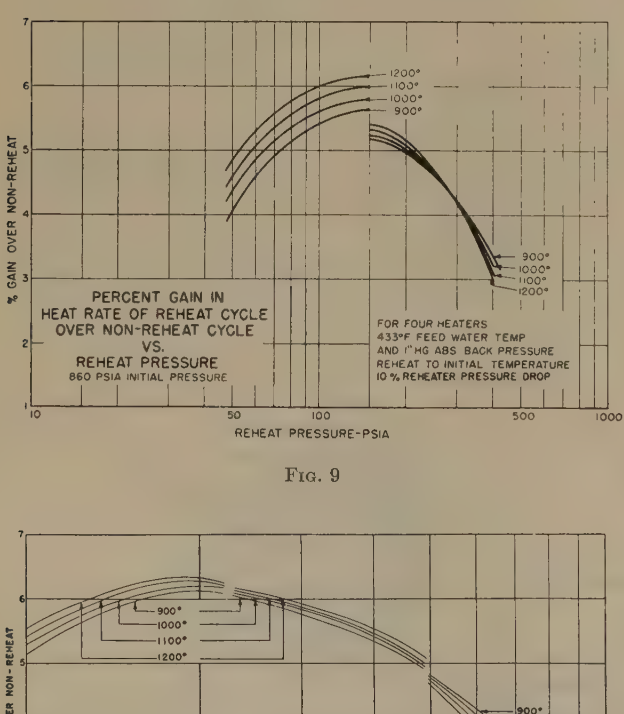

Fig. 9

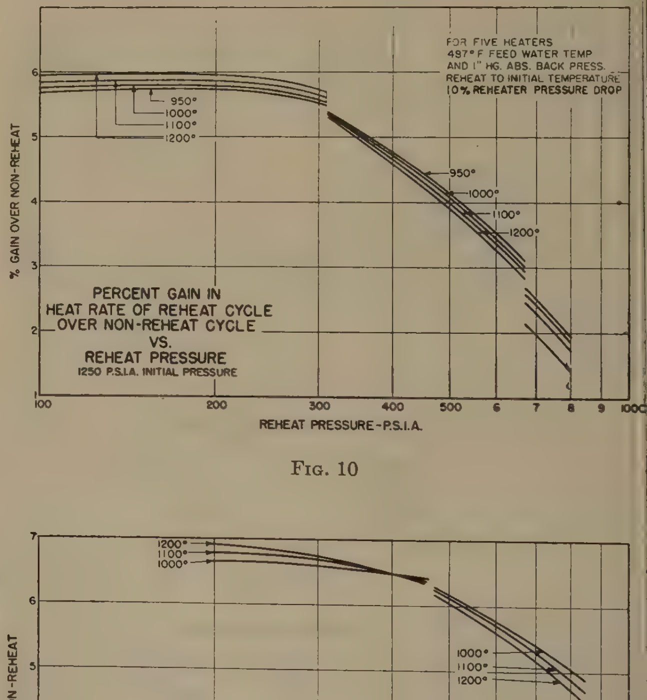

Fig. 10

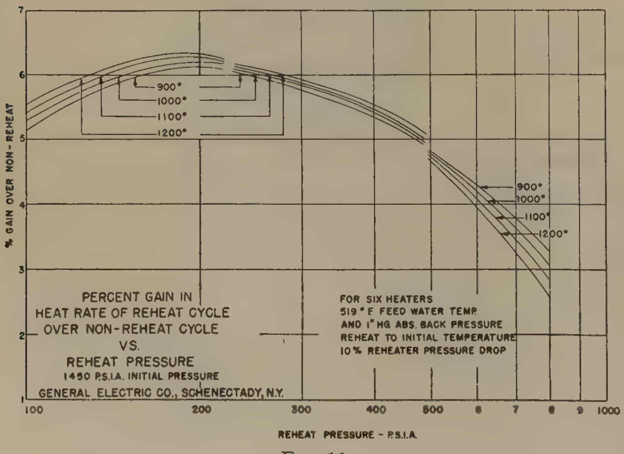

Fig. 11

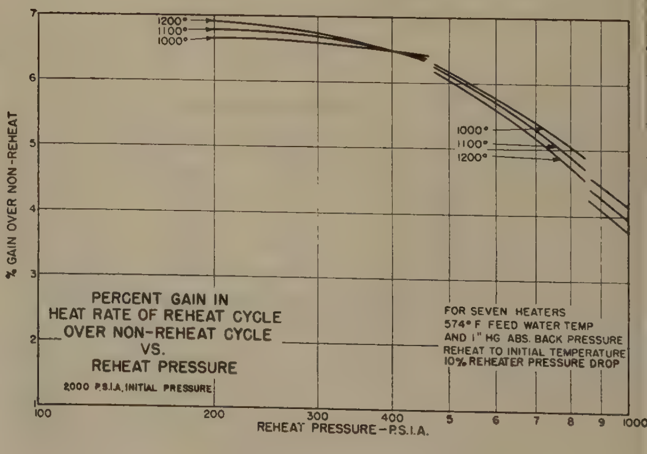

Fig. 12

plotted against per cent change in high-pressure turbine efficiency.

For the nonreheat unit, the corresponding reduction in heat rate for a change in the turbine efficiency down to a pressure corresponding to the reheat pressure on the reheat machine is about 0.17 per cent for each 1 per cent increase in the turbine efficiency. Only the 1450-lb 1000 F unit was calculated, and the range of the variation is shown in Fig. 14.

The difference between the reheat and the nonreheat heat rates also changes with high-pressure-turbine efficiency. This gain in

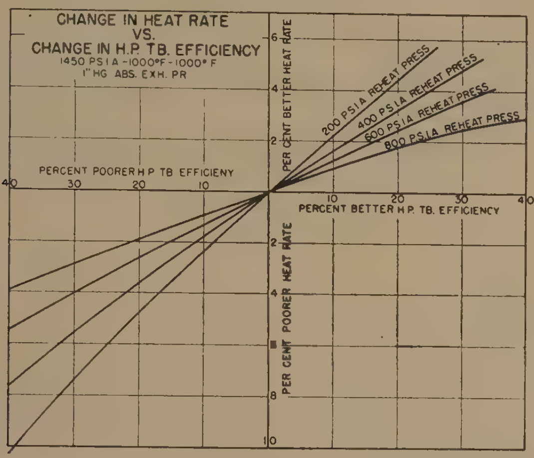

Fig. 13

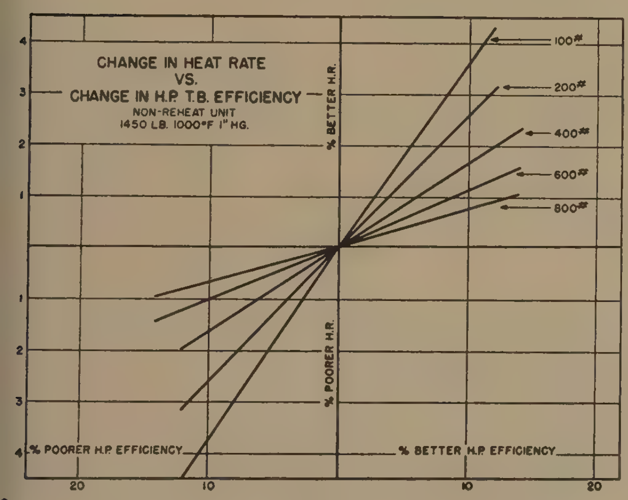

Fig. 14

heat rate for equal efficiencies in the turbines down to the reheat pressure point for the reheat and nonreheat units is plotted against high-pressure turbine efficiency down to this point for four reheater pressures at 1450 lb, 1000 F, 1 in. Hg. in Fig. 15. Within the usual range of reheat pressures the variation in the gain for normal variations in turbine efficiency is not very large. While a complete exploration of this effect with all the variables was not made, the foregoing curves are believed to be representative of all normal combinations.

The heat rate of a reheat cycle is also markedly affected by the pressure drop in the reheater and connecting piping, since such pressure drop is a throttling loss. The basic curves, Figs. 9, 10, 11, and 12, were based upon 10 per cent pressure drop from the high-pressure turbine-outlet flange to the flange of the intercepting valve after the reheater. Figs. 16, 17, and 18 show per cent variation in heat rate with reheater pressure drop.

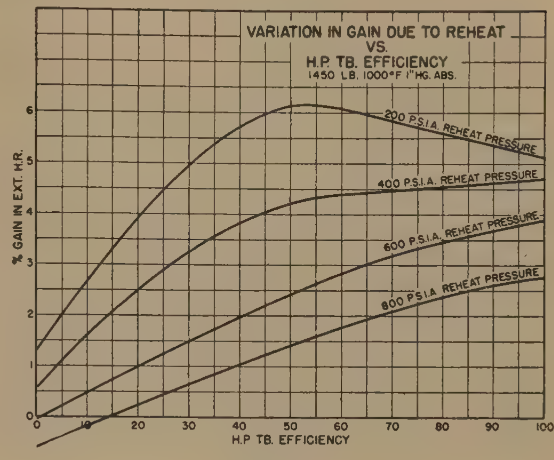

Fig. 15

The effect of the condenser pressure upon the gain due to reheat is rather small, but definite. The principal effect is a reduction in the gain due to leaving loss with constant annulus area, that is, with higher back pressures the exhaust loss of a nonreheat unit is reduced so that the gain due to the decrease in exhaust loss for the reheat machine is also reduced. This effect for units with the same last stage reduces the gain due to reheat by from 0.25 to 0.5 per cent in going from 1 in. Hg abs to $1^1/_2$ in. Hg abs exhaust pressure. Figs. 19, 20, 21, and 22 show a comparison of the gain at 1000 F for 1 in. and $1^1/_2$ in. Hg abs.

The gain due to reheat also varies with load on the unit, since a change in load means a change in flow, with a corresponding change in reheat pressure and feedwater temperature; while a change in flow also changes the efficiency of the high-pressure section. The combined effect of these three variables has not been investigated fully, but the effect is such that the gains at partial loads are less than would be indicated for the reheater pressure corresponding to the load at which the unit is operating. This means that the reheat pressure at rated load should be chosen higher than the optimum for the steam conditions involved in order to maintain economy at partial load.

All the data presented in this paper have been based on reheating to the initial temperature. In some cases the reheat temperature is chosen less than the initial temperature, particularly where the initial temperature is high as this reduces the problems associated with the design of the reheater and low-pressure section of the turbine.

A difference of 50 deg F between the initial and reheat temperatures changes the heat rate about 0.6 per cent for usual temperature ranges.

Conclusions

Reheat has been in use for over 25 years, and there have been few, if any, difficulties attributable to reheat as such with the plants in all parts of the country which are using reheat.

The chief objection to its use in the past has been on the basis of cost, which sometimes overbalanced the gains in fuel costs. Modern developments in turbines and boilers have reduced the differential in cost between reheat and nonreheat units, and steadily increasing fuel costs have emphasized the search for improved economy and enabled the higher costs of a reheat installation to be justified economically in most cases.

As has been shown by the curves presented herein, when due allowances are made for usual range of reheat pressures and high-pressure section efficiency, reductions in heat consumption, and of course correspondingly in fuel consumption and fuel costs, of

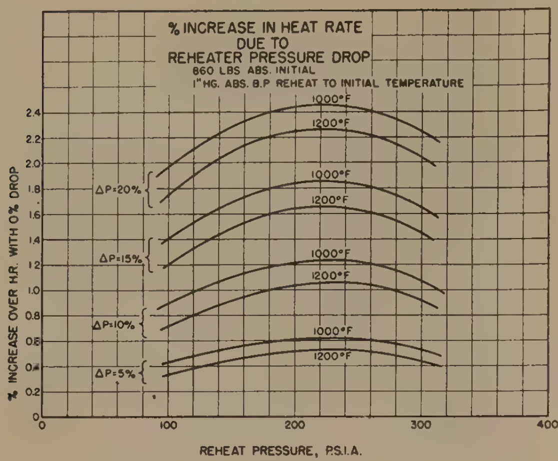

Fig. 16

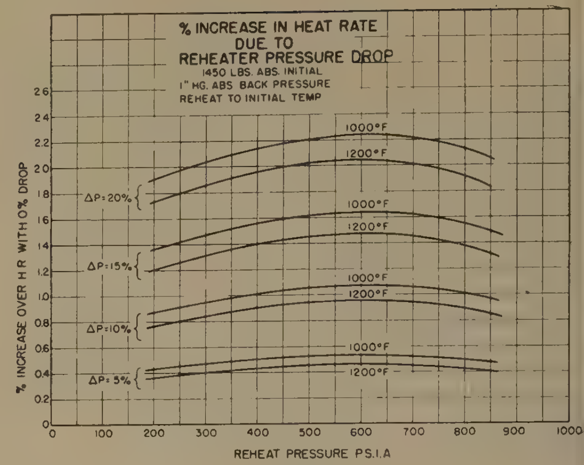

Fig. 17

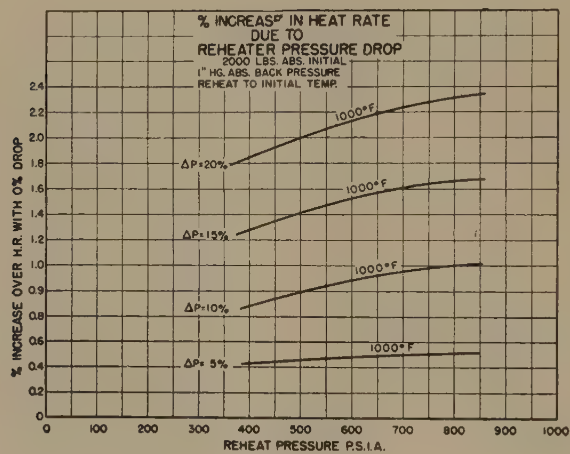

Fig. 18

$4^1/_2$ to 5 per cent may be realized at loads from about $^3/_4$ to $^5/_4$ load. This applies to the higher ratings and moderately high steam pressure. For ratings of 40,000 kw and below, or for very high steam conditions, the lower high-pressure section efficiency attainable with reasonable cost, reduces this gain to from $3^1/_2$ to 4 per cent.

Studies by a number of utilities have shown that the gains indicated in this paper provide an adequate return on the additional investment. It is hoped that such economic studies will be made available to the industry for its guidance.

The industry as a whole has shown an average increase in initial temperature of about 12 deg F per year, and there are no indications that this rate will change in the future. While the increasing application of reheat may not change this trend, it does provide a marked increase in the rate at which plant thermal efficiency has risen over the years and a corresponding marked reduction in that significant ratio, pounds of coal per kilowatthour.

We have every confidence that the revival of interest in, and use of reheat is a permanent one, and that an increasing number of reheat installations will be made in the future. The obvious trend in coal costs is pretty good assurance of this.

Acknowledgment

In closing the authors wish to acknowledge the valuable assistance of Mrs. Jean Higley and Miss Katherine Edwards who made the majority of the calculations on which the curves and data the paper were based.

Appendix

The change in turbine efficiency with superheat mentioned the text was given as Fig. 6, in Warren and Knowlton's paper (This has been extended to higher superheats for use with reh turbines, and in addition, curves have been drawn to include effect of differences in pressure at the same superheat. Th correction curves, drawn as multiplying factors, are given in F 23.

These correction factors are based on calculation supported tests on a large number of turbines and stage groups. They be applied to impulse-type turbines with interstage moist drainage as built by the author's company.

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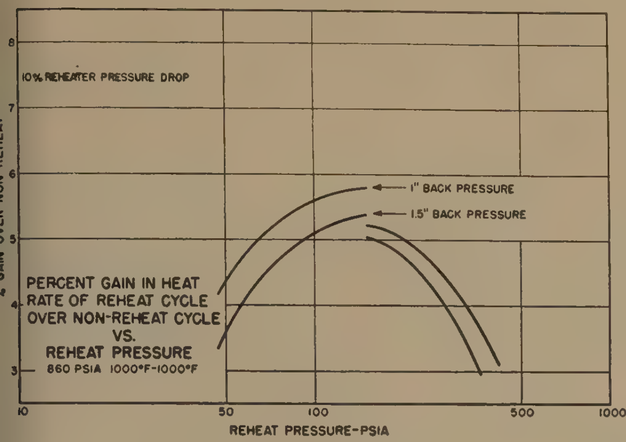

FIG. 19

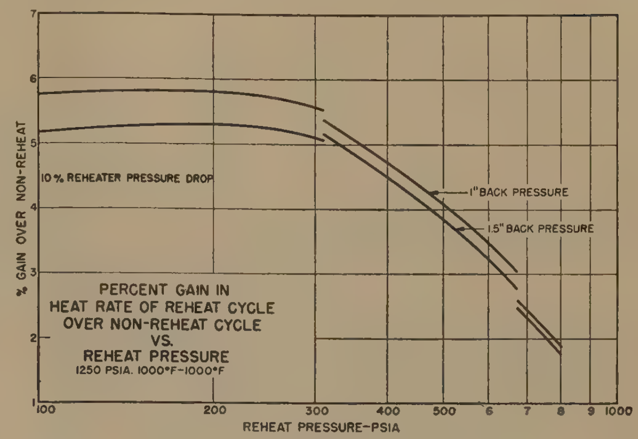

FIG. 20

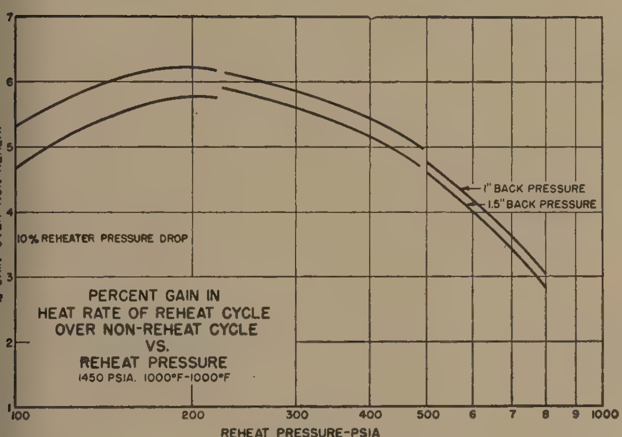

FIG. 21

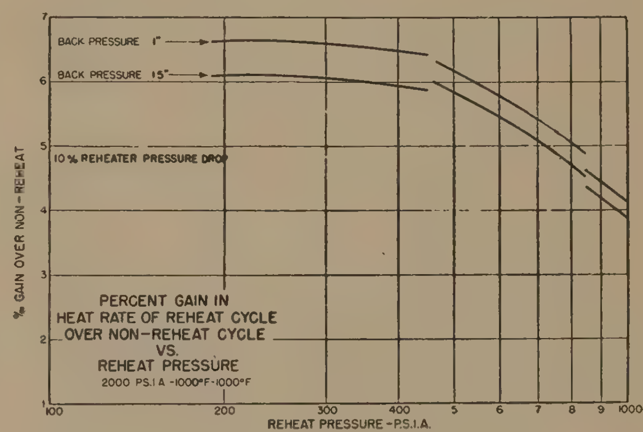

FIG. 22

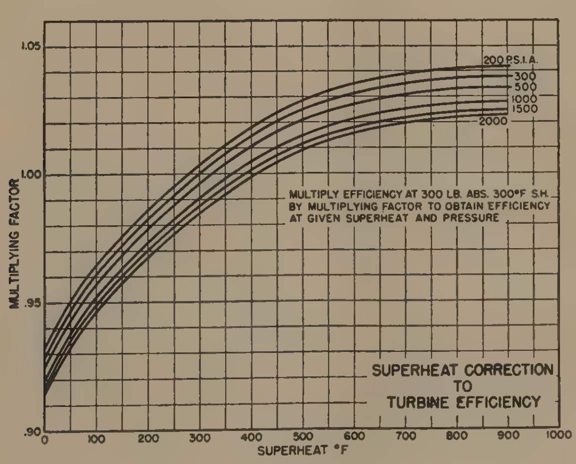

FIG. 23

3 "Resuperheating in Steam Turbines," by W. E. Blowney and . B. Warren, *Mechanical Engineering*, vol. 47, 1925, pp. 455–460.

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Steam Turbines for Resuperheat Cycle

By E. E. Parker,[1] Schenectady, N. Y.

Resuperheat cycles are again being applied to central-station steam-turbine installations; this time at initial and reheat temperatures in the range of 950 to 1050 F. This paper reviews the steam turbines installed by the author's company for the resuperheat cycle and describes the modern turbines under construction for this cycle.

Introduction

In the middle 1920's and early 1930's, a number of installations of steam power generating equipment using the resuperheat cycle were installed for central-station service. This was a period when initial temperatures had reached the 750 F level. Progress to higher initial temperatures was yet a few years in the future. During this period, while experience was being consolidated at the 750 F level, higher-thermal-efficiency stations were built using the resuperheating cycle. Roughly, a thermal gain of 5 to 6 per cent was obtained by using the resuperheating cycle. This gain was not equaled by nonreheat cycles until progress permitted initial temperatures of the order of 900 F. Thus the efficiency of these early resuperheat installations equaled the efficiency of nonreheat installations made years later at higher temperature levels.

Today, initial temperatures of 900 and 950 F are common for central-station service, and a large amount of capacity is being installed at 1000 and 1050 F. It appears likely that operating experience with materials at 1000 and 1050 F will characterize the progress in central-station trends during the next few years. Such consolidation of experience need not, however, at the moment impede the progress in construction of stations for higher thermal efficiency. By application of the resuperheat cycle at 1000 F, plant heat rates can be achieved which in all probability will equal those possible in the future at initial temperatures of 1150 or 1200 F without the use of resuperheat. Thus the present renewed interest in and use of the resuperheat cycle parallels in many respects the conditions that led to its application 25 years ago.

Available records indicate that there is installed in the United States approximately 2,460,000 kw in turbine capacity for the interstage resuperheat cycle. In addition, available records show there is installed some 120,000 kw in noncondensing-turbine capacity which supplies steam to older low-pressure units with provisions for resuperheating the steam between the exhaust of the newer noncondensing turbines and the inlet to the older low-pressure units. This paper is limited to consideration of those turbines built for interstage resuperheat, that is, turbines built to utilize the steam from the throttle to the condenser.

The company with which the author is associated has manufactured 28 turbines with combined capacity of approximately 1,945,000 kw for the interstage resuperheat cycle. Twenty-six of the turbines (see Table 1) are installed in the United States and two with a combined capacity of 105,000 kw in South America. Of this total, units of approximately 300,000 kw use steam resuperheating, the remainder furnace resuperheating. With the exception of one unit, these were all manufactured during the period from 1924-1932.

An indication of the renewed interest in the resuperheat cycle can be gained from the fact that this company had under construction and on order at the middle of 1948, approximately 1,500,000 kw at name-plate rating for the interstage resuperheat cycle (see Table 3). The first of these units will be placed in service in 1949. This renewed interest in the resuperheat cycle is one of many examples of the effort of the electric-utility industry to prevent power costs from rising in the face of rapidly rising fuel and other costs. It is also a contribution to the conservation of our fuel reserves which is assuming increasing importance.

Earlier Designs

Before describing modern turbines now under construction for the resuperheat cycle, it is of interest to note briefly the designs of some of the earlier turbines, built for this cycle, which have been proved by years of successful operation.

Fig. 1 is a cross section of one of the first resuperheat turbines built by the author's company. A number of turbines of this type are in operation, and these are of particular interest as the resuperheating is here applied to a single-casing turbine, giving the advantages of compactness for station design and minimum capital investment required for resuperheating. This design, however, is subjected to large temperature gradients in the section of the turbine comprising the exhaust to the reheater and inlet from the reheater. Such gradients are not as serious in these units designed for 750 F as would exist at today's temperature levels.

The second type of construction used for early resuperheat turbines is the tandem compound of which there is a variety of combinations. Fig. 2 shows an 1800-rpm tandem-compound double-flow three-casing unit with resuperheating between the high and intermediate sections. This is one of the later units of the group now in service and operates at steam conditions of 1200 psig, 825 F initial temperature, with resuperheat to 825 F at a maximum pressure following the reheater of 425 psig.

Many of the early resuperheat turbines are of cross or vertical-compound design. The high-pressure turbines of such sets are in some cases 1800 rpm and others 3600 rpm. The low-pressure turbines are 1800 rpm, either single or double flow.

Fig. 3 shows the high-pressure and Fig. 4 the low-pressure turbine of a vertical-compound turbine. These cross sections also serve as typical of cross-compound designs. The high-pressure 3600-rpm turbine and generator of this vertical-compound unit are mounted on the generator for the low-pressure turbine. The low-pressure turbine is single flow and operates at 1800 rpm. Turbines such as this were built for steam conditions of 1200 psig 750 F initial temperature, with resuperheating to 750 F. The steam is resuperheated between the high and low-pressure turbines at a maximum reheat pressure of 450 psig.

A cross-compound unit for operation at 2300 psig initial pressure, installed in the Twin Branch Station of the Indiana and Michigan Electric Company,[2] is the most recent installation of resuperheating turbines manufactured by the author's company.

[1] Divisions Engineer, Steam Turbine and Generator Divisions, General Electric Company. Mem. ASME.

Contributed by the Power Division and presented at the Annual Meeting, New York, N. Y., November 28–December 3, 1948, of The American Society of Mechanical Engineers.

 Paper No. 48—A–58.

[2] Series of articles in *Electrical World*, Oct. 18, 1941.

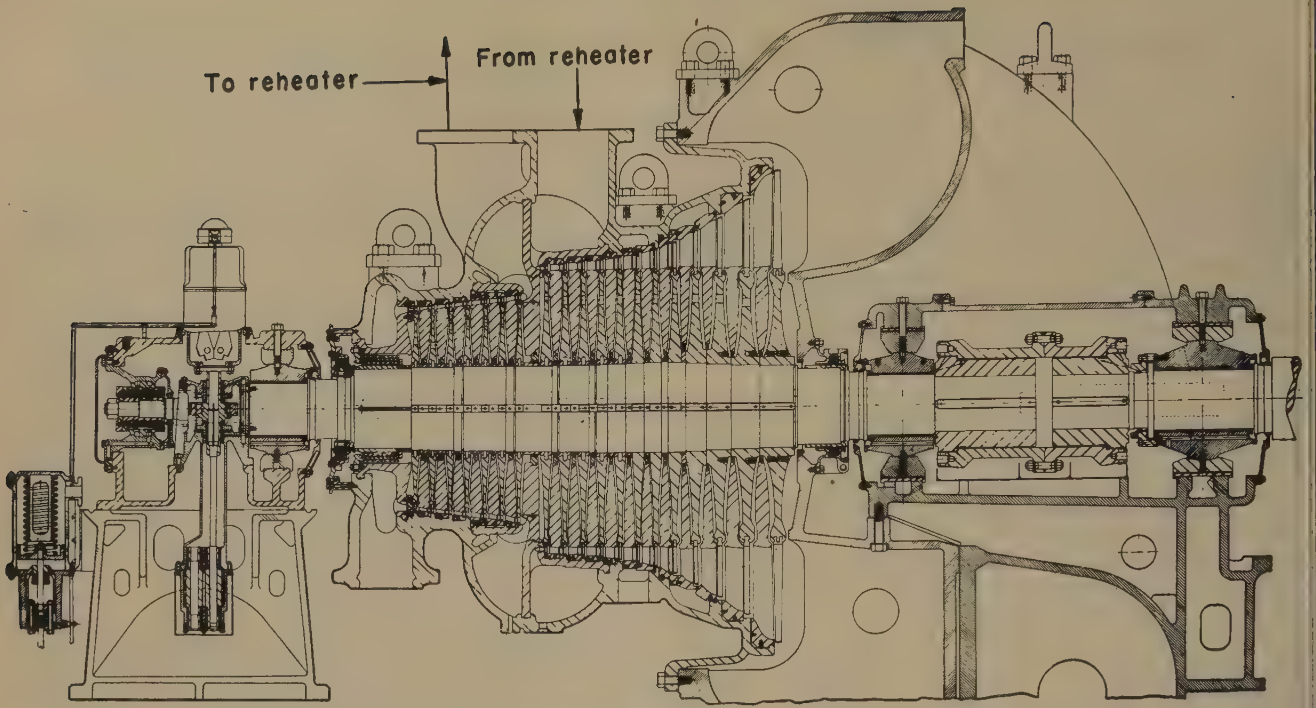

Fig. 1 35,000-Kw 1800-Rpm Single-Cylinder Steam Turbine of Early Design for Resuperheat Cycle

TABLE 1 LIST OF RESUPERHEAT STEAM TURBINES OF THE AUTHOR'S COMPANY, WITH INTERSTAGE RESUPERHEATING, INSTALLED IN THE UNITED STATES

Purchaser	Rating, kw	Initial steam conditions: Press, psig	Initial steam conditions: Temp, deg F	Reheat conditions: Press inlet, psig	Reheat conditions: Temp, deg F	Type of turbine	Type of reheat	Date in service
A	35000	530	725	134	725	SC	Boiler	Sept. 1924
B	60000	550	725	95	725	CC SF	Boiler	Nov. 1924
A	35000	530	725	134	725	SC	Boiler	Feb. 1925
C	35000	530	725	134	725	SC	Boiler	April 1925
C	35000	530	725	134	725	SC	Boiler	July 1925
D	63000	600	725	187	725	CC SF	Boiler	Nov. 1925
D	63000	600	725	187	725	CC SF	Boiler	March 1926
E	42000	600	725	160	725	SC	Boiler	March 1927
E	42000	600	725	160	725	SC	Boiler	June 1927
B	90000	550	750	75	460	CC SF	Steam	Aug. 1927
F	55000	600	725	155	725	TC SF	Boiler	Aug. 1928
A	165000	600	725	117	725	CC	Boiler	Jan. 1929
G	208000	600	730	100	500	CC	Steam	April 1929
B	55000	600	725	155	725	TC SF	Boiler	June 1929
H	55000	1250	750	342	750	CC SF	Boiler	Feb. 1930
J	53000	1200	725	390	750	CC SF	Boiler	March 1930
J	53000	1200	725	390	750	CC SF	Boiler	May 1930
K	25000	1250	750	320	750	VC SF	Boiler	Sept. 1930
K	25000	1250	750	320	750	VC SF	Boiler	Oct. 1930
F	105000	600	725	158	750	TC DF	Boiler	Nov. 1930
L	50000	1200	750	425	750	VC SF	Boiler	Feb. 1931
L	50000	1200	750	425	750	VC SF	Boiler	June 1931
M	110000	1200	750	73	550	VC DF	Steam	July 1931
G	150000	1200	825	410	825	TC DF	Boiler	Nov. 1937
C	76500	2300	940	430	900	CC SF	Boiler	March 1940
B	105000	650	750	158	750	TC DF	Boiler	May 1940

Notation:
SC—Single cylinder.
CC SF—Cross compound, single flow.
TC SF—Tandem compound, single flow.
TC DF—Tandem compound, double flow.
VC SF—Vertical compound, single flow.
VC DF—Vertical compound, double flow.

TABLE 2 LIST OF NONCONDENSING STEAM TURBINES OF THE AUTHOR'S COMPANY IN THE UNITED STATES, WITH RESUPERHEATING OF EXHAUST STEAM

Purchaser	Rating, kw	Initial steam conditions: Press, psig	Initial steam conditions: Temp, deg F	Reheat conditions: Press, exhaust, psig	Reheat conditions: Temp, deg F	Date in service
N	3360	1200	700	350	750	Dec. 1925
O	7700	1200	750	300	750	Oct. 1926
N	10000	1200	700	375	750	Dec. 1927
N	12500	1200	700	375	750	Oct. 1929
O	7700	1200	750	317	750	Nov. 1929
O	7700	1200	750	317	750	Nov. 1930
O	7700	1200	750	317	750	Sept. 1930
P	12000	1282	710	375	685	Feb. 1932
R	15000	1250	800	310	687	Sept. 1938
S	20000	1800	950	400	750	Sept. 1942

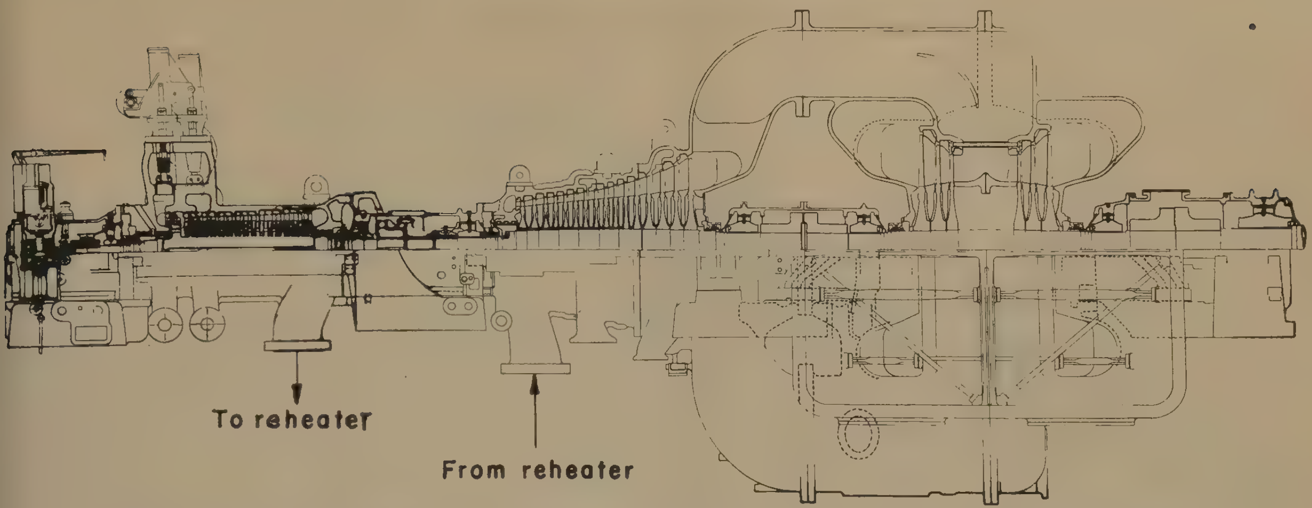

FIG. 2 150,000-KW 1800-RPM THREE-CASING TANDEM-COMPOUND STEAM TURBINE OF EARLY DESIGN FOR RESUPERHEAT CYCLE

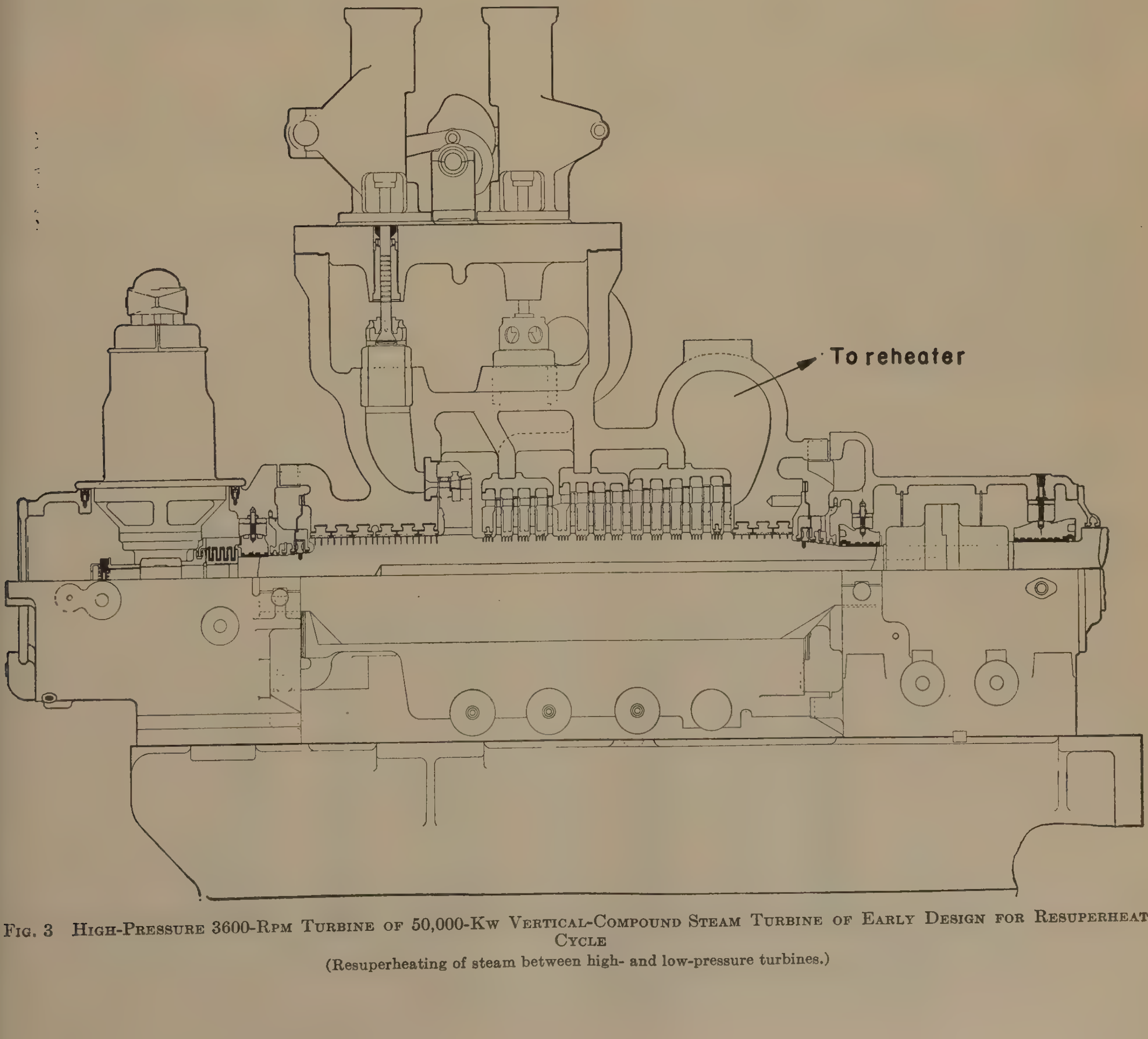

FIG. 3 HIGH-PRESSURE 3600-RPM TURBINE OF 50,000-KW VERTICAL-COMPOUND STEAM TURBINE OF EARLY DESIGN FOR RESUPERHEAT CYCLE

(Resuperheating of steam between high- and low-pressure turbines.)

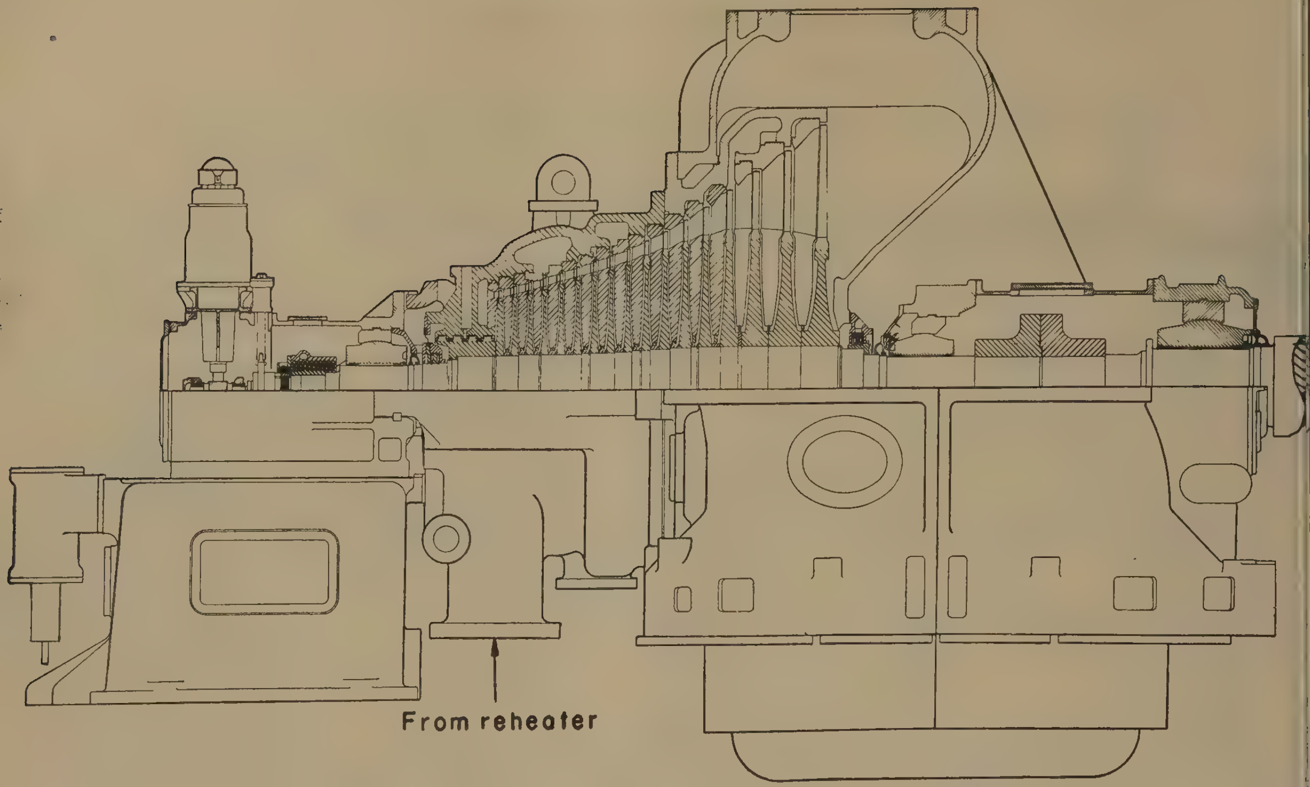

Fig. 4 Low-Pressure 1800-Rpm Turbine of 50,000-Kw Vertical-Compound Steam Turbine of Early Design for Resuperheat Cycle

This installation provides resuperheating between the high and low-pressure turbines using 940 F initial steam temperature, with resuperheating of the steam to 900 F. A station heat rate of 10,035 btu per kwhr has been reported on this resuperheat installation for 6 months of operation.[3] Up to July, 1948, no record of station heat rate on a steam-electric installation has been reported which equals this level of thermal efficiency.

Current Designs

Two types of resuperheat turbines tandem and cross-compound, and varying in rating from 50,000 to 125,000 kw are currently being manufactured by the author's company for resuperheat installations now under construction by the electric-utility industry.

Representing the cross-compound type are a number of resuperheat units for installation in the Philip Sporn[4] and other stations of the American Gas and Electric Company. Details of these 125,000-kw turbines (150,000 kw maximum capacity) are shown in Fig. 5 for the high-pressure, and Fig. 6 for the low-pressure units. The 3600-rpm high-pressure turbine receives steam at 2000 psig and 1050 F, and exhausts it to the reheater where it is resuperheated to 1000 F. The steam then enters the 1800-rpm tandem-compound double-flow low-pressure turbine. At maxi-

[3] "Operating History of the 2500-Psi Twin Branch Plant," by Philip Sporn and E. G. Bailey, Trans. ASME (Special Section—Furnace Performance Factors), vol. 66, 1944, pp. 1–12.

[4] "The 2000-Psi, 1050 F, and 1000 F Reheat Cycle at the Philip Sporn and Twin Branch Steam-Electric Stations," by Philip Sporn, Trans. ASME, vol. 70, 1948, pp. 287–294.

TABLE 3 LIST OF RESUPERHEAT STEAM TURBINES FOR INTERSTAGE RESUPERHEATING, ON ORDER WITH THE AUTHOR'S COMPANY AUGUST 1, 1948

Purchaser	Rating, kw	Initial steam conditions: Press, psig	Initial steam conditions: Temp, deg F	Reheat conditions: Press, TB inlet, psig	Reheat conditions: Temp, deg F	Type of turbine	Type of reheat
C	125000	2000	1050	375	1000	CCDF	Boiler
T	125000	2000	1050	375	1000	CCDF	Boiler
A	125000	2000	1050	375	1000	CCDF	Boiler
C	125000	2000	1050	375	1000	CCDF	Boiler
T	125000	2000	1050	375	1000	CCDF	Boiler
A	125000	2000	1050	375	1000	CCDF	Boiler
U	80000	1450	1000	375	1000	TCDF	Boiler
U	80000	1450	1000	375	1000	TCDF	Boiler
U	80000	1450	1000	375	1000	TCDF	Boiler
V	60000	1450	1000	375	1000	TCDF	Boiler
V	60000	1450	1000	375	1000	TCDF	Boiler
W	50000	1450	1000	405	1000	TCDF	Boiler
X	60000	1450	1000	355	1000	TCDF	Boiler
Y	50000	1450	1000	405	1000	TCDF	Boiler
Y	50000	1450	1000	405	1000	TCDF	Boiler
Z	100000	1250	950	380	950	TCDF	Boiler
Z	100000	1250	950	380	950	TCDF	Boiler

Notation:
CCDF—Cross-compound, double-flow.
TCDF—Tandem-compound, double-flow.

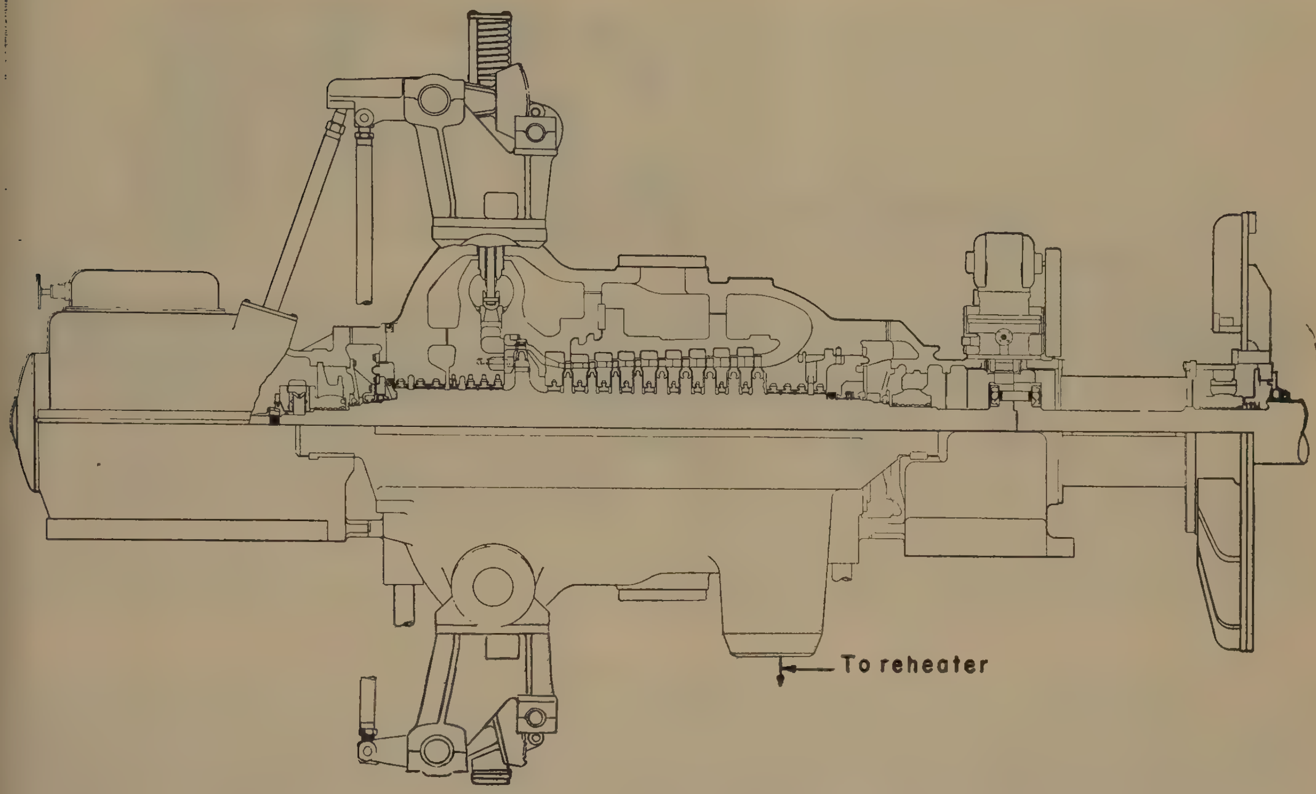

FIG. 5 2000-PSI 1050 F 35,000-KW 3600-RPM HIGH-PRESSURE TURBINE OF 125,000-KW CROSS-COMPOUND TURBINE FOR RESUPERHEAT CYCLE

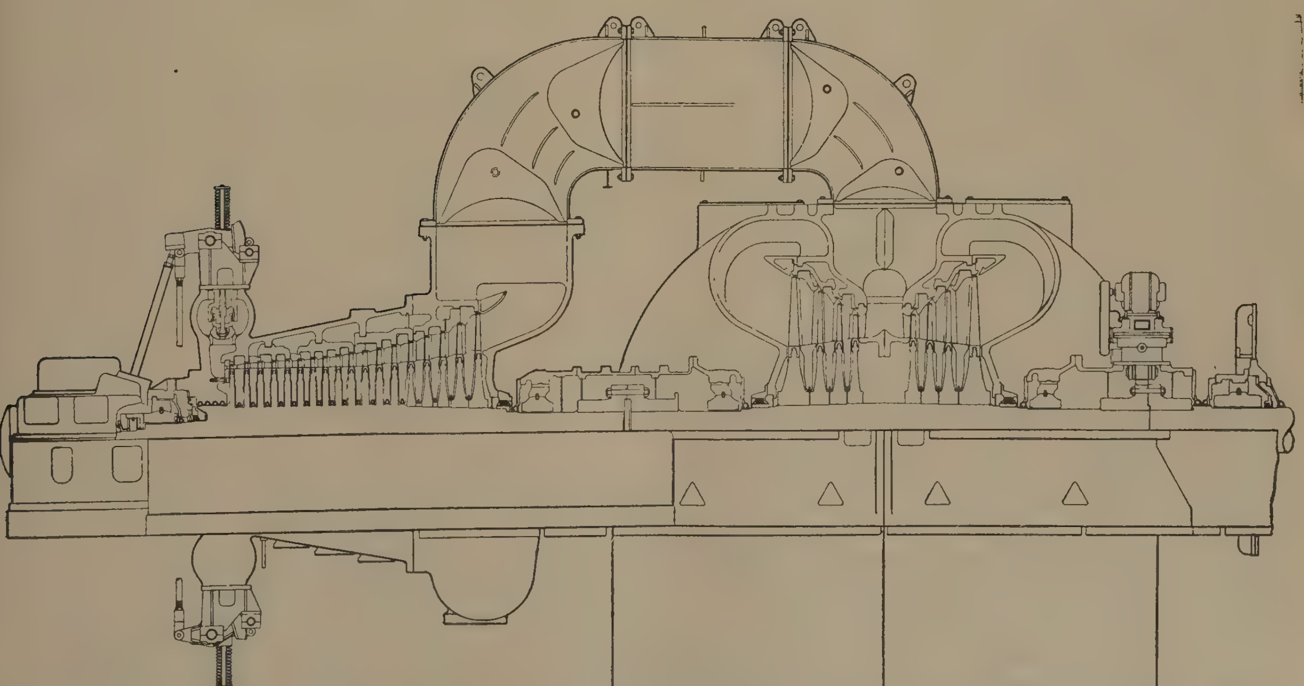

FIG. 6 90,000-KW 1800-RPM TANDEM-COMPOUND DOUBLE-FLOW LOW-PRESSURE TURBINE OF 125,000-KW CROSS-COMPOUND TURBINE FOR RESUPERHEAT CYCLE
(Receives steam from reheater at 1000 F.)

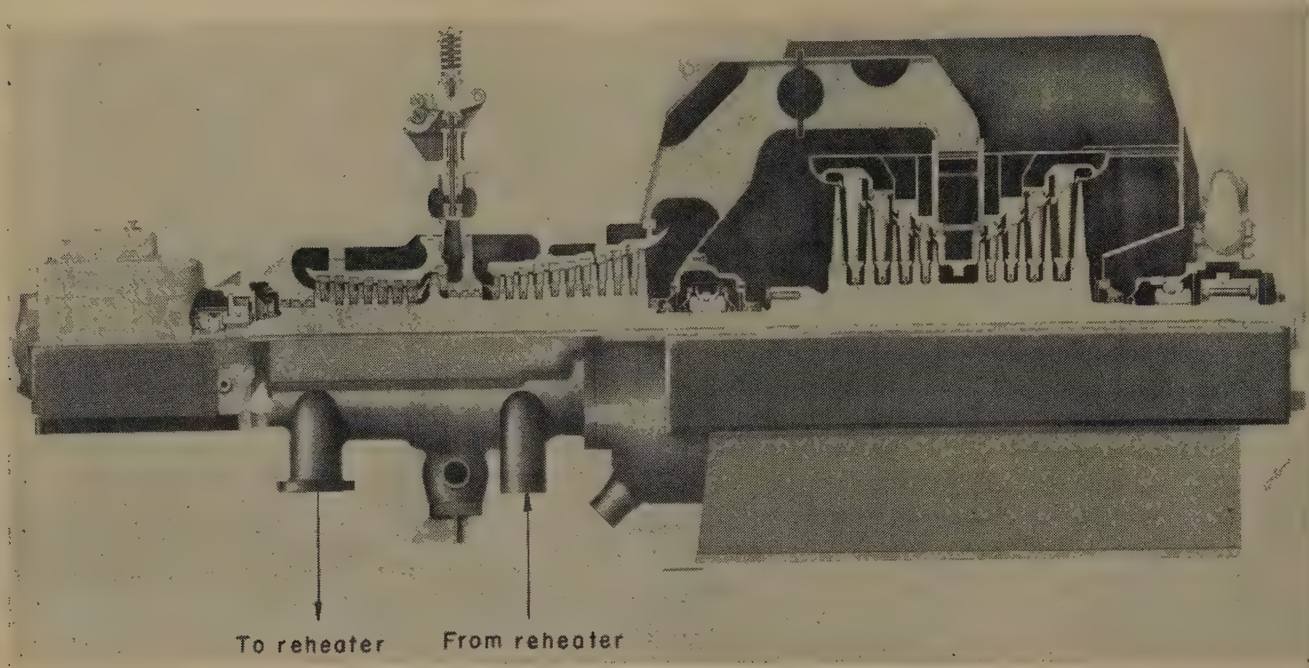

Fig. 7. Recent Development in Steam Turbines for Resuperheat Cycle
(80,000-kw 3600-rpm tandem-compound double-flow turbine.)

mum load, the steam pressure entering the low-pressure turbine is 375 psig.

The high-pressure turbine has many special features to make it suitable for the high initial pressure and temperature conditions, among which are special stabilized stainless-steel materials for all parts operating at 1050 F, double high-pressure shell construction,[5] and a special arrangement of packing steam flow to cool the outer shell. Because of the double-shell construction, the use of stainless steel for the turbine shell can be limited to the inner shell, and only the forward portion of it extending beyond the second stage. At that point the stainless-steel portion is dovetailed to a special low-alloy-steel shell.

The 1800-rpm low-pressure turbine is a two-casing tandem-compound unit with a double-flow low-pressure section. This double-flow section provides adequate last-stage area for a turbine of this size at even the best vacuum conditions. It is the first large 1800-rpm turbine to be designed for 1000 F service, and the high-pressure section employs a solid rotor in the high-temperature region, combined with built-up wheel construction for the later stages which operate at lower temperatures. The intercept valves are located in the high-pressure shell of the low-pressure unit and are operated by the same arrangement of mechanism as normally used for governor-operated control valves.

A recent development in turbines for the resuperheat cycle is shown in Fig. 7. This is a 3600-rpm tandem-compound double-flow turbine which resembles very closely the equivalent-size nonreheat turbine. In this design the high-pressure steam enters the turbine shell at mid-section and travels forward (away from the generator). It is exhausted to the reheater at the forward end of the turbine adjacent to the No. 1 bearing standard and re-enters from the reheater immediately adjacent to the point at which the high-pressure steam is admitted. Thus only one portion of the turbine shell is subjected to the high temperature of the initial and resuperheated steam. This eliminates any severe temperature gradients, as would exist in the earlier, compact design shown in Fig. 1. It also moves the highest-temperature portion of the turbine shell and shaft away from the bearings and water seals.

Foundation and station space required for this type unit are essentially the same as for a unit of similar rating without resuperheating. This, together with the fact that the turbine does not require an additional cylinder to accommodate resuperheating, minimizes the capital investment to build a station using this cycle.

Turbines of this type are under construction in ratings of 50,000 kw, 60,000 kw, 80,000 kw, and 100,000 kw with steam conditions of 1450 psig, 1000 F initial temperature and reheat to 1000 F, or 1250 psig, 950 F initial, and reheat to 950 F. This type of resuperheat turbine can be designed for other steam conditions and may be applied to any existing size condensing unit for which the resuperheating cycle can be justified economically. To make possible 3600-rpm turbines of larger rating, or for improved vacuum conditions, a triple-flow low-pressure section instead of a double-flow section, as shown in Fig. 7, can be applied to this type of resuperheating turbine equally as well as to nonreheat turbines.

Control Requirements

A turbine for the resuperheat cycle requires those controls customarily used on a nonreheat turbine plus an intercept valve, together with its control. Referring to Fig. 8, this intercept valve is located close to the turbine in the line returning the steam from the reheater. The sole purpose of this intercept valve is to prevent the turbine speed from reaching 110 per cent, and thus tripping the emergency governor when a sudden loss of all or all but a small portion of the electrical load occurs. If it were not for the intercept valve, the unit, upon loss of electrical load, would go up in speed and trip the emergency stop valve, even though the main steam valves were closed, owing to the continuing flow of the steam stored in the reheater and associated piping through the low-pressure section of the turbine.

The intercept valve is controlled by a speed governor termed a

[5] "Progress in Design and Performance of Modern Large Steam Turbines for Generator Drive," by G. B. Warren, Trans. ASME, vol. 63, 1941, pp. 49–75.

pre-emergency governor, which is designed to operate the valve through full stroke with a change in speed of approximately 3 per cent. It is normally set so that the valve is fully open at all speeds below about 102 per cent. At a speed of 102 per cent, the valve starts to close and is fully closed at a speed of 105 per cent. If the intercept valve closes due to a rise in turbine speed, it will reopen upon reduction of speed as it is a balanced-type valve.

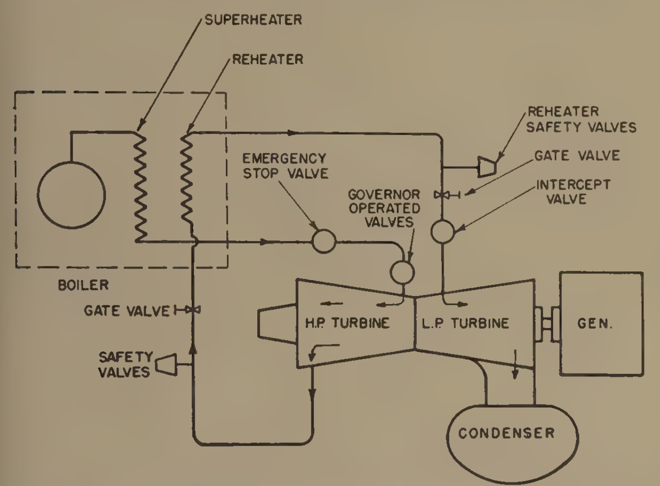

Fig. 8 Diagram Showing Location of Intercept Valve and Safety Valves in Reheater Steam Lines

Because of the addition of the intercept valve, full-capacity safety valves must be installed between the exhaust of the high-pressure turbine and the intercept valve. These valves, if located on the high-temperature side of the reheater, are normally set to start relieving approximately 10 per cent above the maximum pressure at the intercept valve. For hydrostatic tests of the reheater, gate valves, as shown in Fig. 8, or removable flanged sections to permit blanking off the reheat lines, must be provided. If gate valves are used, safety valves must also be provided as shown in Fig. 8, to protect the exhaust of the high-pressure turbine and a portion of the reheater piping.

Fig. 9 illustrates in some detail a schematic control arrangement for a modern resuperheat turbine.

Although only an intercept valve and a governor for its operation need be added to the usual turbine controls for a resuperheat turbine, the operation and control of a resuperheat installation involve more problems than this alone would indicate. There are two major problems to be solved on each resuperheat installation, namely: (1) protection of the reheater upon sudden loss of load or during starting; and (2) protection of the exhaust portion of the turbine and the condenser against overheating during starting or operation at no load or light loads of a few thousand kilowatts.

The first of these problems, that is, protection of the reheater, is being met on modern installations by automatic provision for controlling the fires when all, or all but a small portion of the turbine load, is suddenly lost. By adequate indicating means, the boiler operator can be informed of the load conditions on the turbine

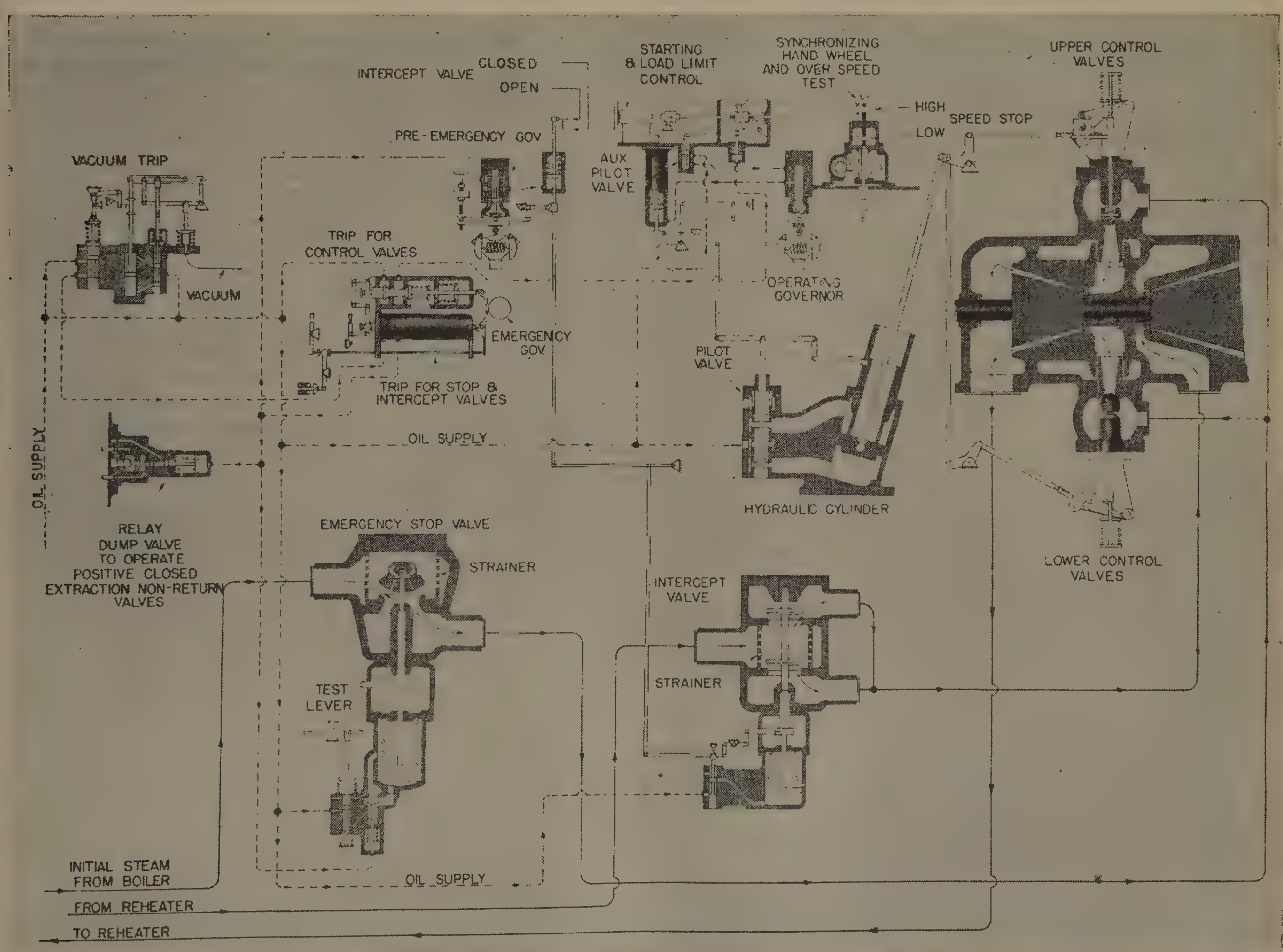

Fig. 9 Schematic Diagram of Controls for a Resuperheating Turbine

following the loss of load, from which the subsequent operation of the boiler can be determined.

The protection required to prevent overheating of the exhaust end of the turbine depends upon the boiler characteristics and the operating cycle. For short periods of operation at no load or light loads, it is probable that no protection will be found necessary. For prolonged periods of operation at no load or light loads, provision for protection should be made if the characteristics of the boiler are such that there is a temperature rise of any magnitude in the steam passing through the reheater. At light loads, after steady conditions have been reached, it is a reasonable approximation to assume that the exhaust of a resuperheat turbine will operate hotter than a similar nonreheat turbine by the number of degrees temperature rise of the steam passing through the reheater, provided vacuum and temperature conditions are such that the exhaust of the nonreheat unit is saturated or superheated. Thus a temperature rise of 50 deg F through the reheater may cause the exhaust end of a resuperheat turbine to operate at no load approximately 50 deg F higher than the exhaust of a similar nonreheat unit supplied steam at the same initial conditions. An increase in exhaust temperature of even this magnitude is considered undesirable. To overcome this condition, desuperheating of the steam is required. This may be accomplished by a desuperheater in the exhaust from the reheater or in the turbine crossover just ahead of the low-pressure turbine. Such a desuperheater would be operated during prolonged operation at no load or very light loads. Desuperheating immediately following the reheater has the disadvantage of subjecting the reheater piping returning from the boiler and the high-temperature portion of the turbine following the reheater to more temperature changes than otherwise would exist. Desuperheating of the steam in the turbine crossover during no-load or light-load operation does not have this disadvantage, and therefore is provided on all resuperheat turbines being manufactured by the author's company.

If other conditions permit, the turbine may be started at reduced boiler pressure and the turbine brought up to speed and partially loaded before the boiler reaches its operating pressure. It is possible that bringing the turbine and boiler up together will eliminate the need for desuperheating the steam from the reheater during starting. This is dependent, however, upon many factors, important among which are the boiler characteristics. These must be studied for each particular installation.

On resuperheat units of the type represented in Fig. 7, it is desirable that the initial and reheat temperatures remain as close to the same value as possible. However, it is to be expected that at light loads, the boiler will not maintain either the initial or reheat temperatures at the rated values, and one or the other of these temperatures will drop off more rapidly, thus increasing appreciably the temperature difference between the initial and reheat steam. Care should be taken in the boiler design to minimize such temperature differences.

Due to the storage of steam in the reheater and its piping, speed governing of a resuperheating unit which is carrying its load isolated from other prime movers is more difficult than a corresponding nonreheat unit. It is felt that this problem can be met satisfactorily for central-station units with the usual amounts of reheater and reheater piping volumes.

If a resuperheat unit, running at full load, should suddenly lose all, or all but a few thousand kilowatts of its load, the speed will rise enough to close the intercept valve, and the unit will continue to operate at a speed sufficiently high to keep the intercept valve closed. Under this condition, high-pressure steam passes through the high-pressure turbine under control of the governor valves and exhausts to atmosphere through the reheater safety valves (see Fig. 8). This condition is corrected by moving the governor speed control down to restore the unit to normal speed. When the speed drops, the intercept valve will reopen and operation becomes normal. Automatic means may be provided to run the governor speed control down upon opening of the generator circuit breaker and thus eliminate or minimize the blowing of the reheater safety valves upon loss of full load. Such resetting of the governor is necessary to resynchronize on either a resuperheat or nonreheat unit when the generator is suddenly electrically disconnected while carrying an appreciable load.

Rate of Loading

In considering a resuperheat installation, the question arises concerning the starting and permissible rate of load change as compared with a nonreheat unit. Considering only the turbine, the time to start and bring a resuperheat turbine to speed should not differ from that for a nonreheat turbine of comparable size and design. Similarly, if desuperheating of the reheated steam at light loads is done, if required, in the turbine crossover, and the control of reheat temperature is as accurate as the control of initial temperature, the maximum rate of load change is the same for a resuperheat as for a comparable nonreheat turbine.

Conclusions

For steam-electric stations, the use of the resuperheating cycle permits the construction of new generating units with station heat rates 4 to 5 per cent lower than nonreheat installations at the same temperature and pressure levels. The return from this improvement in efficiency is appreciable and merits the careful consideration of this cycle for all new installations, particularly with present-day fuel costs.

From a turbine standpoint, the resuperheat cycle not only considerably reduces the number of stages in the moisture region, but also reduces the moisture content in those few stages remaining in this region. The elimination or reduction of moisture on turbine steam-path parts increases the useful life of these parts.

Recent improvements in equipment for this cycle have been made to reduce the capital investment and more completely to insure operating performance comparable to nonreheat units. Studies of the operating experience on existing resuperheat installations, and consideration of the operating problems to be met on the new resuperheat installations under construction have not disclosed any unusual conditions which should impede a continued wide application of this cycle.

Reheating in Steam Turbines

By R. L. REYNOLDS,[1] LESTER, PA.

This paper outlines the history of the reheat cycle, in which steam is resuperheated after expanding through the high-pressure section of the turbine, and shows the improvement in thermal efficiency to be derived from its use under various operating conditions. Such factors as regenerative feedwater heating, pressure drop through the reheater and its piping, and the temperature to which steam is reheated affect the gain in thermal efficiency obtainable with reheating. Reheating reduces throttle and exhaust-steam flows, exhaust moisture, and heat absorbed by the condenser. The effect of all of these factors is illustrated by the curves included in the paper.

THE present upward trend in fuel costs has revived interest in the reheat cycle, since the resulting gain in thermal efficiency will in many cases more than justify the necessary additional cost of station equipment. Growth in power systems and size of individual turbine-generator units further increase the economic possibilities of the reheat cycle.

This paper will describe briefly the history of the reheat cycle, show the advantages to be derived, and point out present and future possibilities.

History

Except for a few isolated and relatively minor cases of reheat, this cycle first came into use about 1925. At that time, materials and turbine construction details limited operating steam temperatures to 700 to 750 F. The use of higher-speed blades in the low-pressure section of condensing turbines simplified the construction and materially reduced space requirements, but in turn limited the exhaust moisture to about 12 per cent because of the water erosion on these low-pressure blades. The importance of this erosion problem was further accentuated by a gradual improvement in turbine stage efficiency, to the end that the initial steam pressure was limited to about 400 psig.

These factors brought about the limited use of the reheat cycle to permit an increase in initial pressure above the 400-psig level.

In order to avoid the increased cost and complexity of the reheat cycle, and at the same time to realize the thermal gain available with higher steam pressures and temperatures, the following improvements in turbine design were made:

(*a*) Improvement in the shape of the low-pressure blades to reduce the effect of erosion.

(*b*) Attachment of erosion-resisting strips on the inlet edges of the rotating blades in the low-pressure section.

(*c*) Modification of low-pressure cylinder design to include water catchers to entrain some of the moisture from the steam passing through the low-pressure blades.

(*d*) Development of alloy-steel materials to permit higher initial steam temperatures.

By these means, the temperatures in the early part of the 1930–1940 decade had risen to about 850 F, and the improvement in the low-pressure blade and cylinder design had made the erosion problem relatively less important. As a result, 600 psig, and occasionally higher pressures, became common, and the reheat cycle was seldom used.

With the revival in central-station-turbine activity during the latter part of the 1930–1940 decade, initial temperatures of 900 F became quite common, permitting the use of 850 psig, and later 1250 psig, initial pressures. This steam temperature level has since increased to 1050 F, accompanied by 1500 psig and higher pressures, in a few cases. However, it is now clear that service experience is needed to prove the dependability and practical economy of power plants constructed for today's advanced steam conditions, temperature in particular, and that this experience should be obtained before making further substantial steps forward. Hence the reheat cycle is experiencing a rebirth of popularity during this period.

Methods of Reheat

Three methods of reheat have been used, as follows:

(*a*) Boiler reheat.
(*b*) Steam reheat.
(*c*) Combination of boiler and steam reheat.

In arrangement (*a*) steam, after expanding through the high-pressure section of the turbine, passes through a fuel-fired reheater in which its temperature is again raised to, or nearly to, the initial temperature, after which the steam expands through the intermediate and low-pressure turbine blading.

In arrangement (*b*) steam is reheated to slightly less than its initial saturation temperature by high-pressure steam.

Arrangement (*c*) combines the two systems, with the steam passing through the steam and fuel-fired reheaters in series.

Reheating with live steam has little or no effect on the heat rate and was used only to reduce exhaust moisture. For this reason this system is no longer being considered.

The combined fuel-fired and steam reheat cycle was used in an effort to maintain closer control of the reheat steam temperature by controlling thermostatically the amount of live steam to the steam-reheater section. This system led to complexity of control and is no longer found necessary.

Thus the only reheat system now being considered is the fuel-fired arrangement, usually with the reheat section being built into the main boiler and interposed between the primary and secondary superheater sections.

Advantages of Reheat

The fuel-fired reheat system has the following advantages:

(*a*) Increase in thermal efficiency, resulting in a reduction in heat and fuel-consumption rates.

(*b*) Reduction in exhaust moisture.

The use of reheat makes it possible to attain high thermal efficiencies without the need for resorting to high initial steam temperatures. For example, when steam is reheated to its initial temperature, the reduction in heat rate is equivalent to that obtained by increasing the initial steam temperature by 150 to 200 F with the nonreheat cycle. Thus a 900 F turbine, reheating to 900 F, will have about the same thermal efficiency as a 1075 F turbine without reheat. This means that the same station heat rate can be obtained with temperatures now considered moderate and well tested as can ultimately be obtained

[1] Manager, Central Station Turbine Section, Westinghouse Electric Corporation. Mem. ASME.

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with temperatures outside the range of present-day central-station experience.

The reduction in exhaust moisture means an improvement in stage efficiency in this section of the turbine and also reduces the erosion on the low-pressure blades. Since this erosion has been reduced by protective strips and by improved blade and cylinder construction, this factor is now relatively less important than when reheat was first introduced.

Effect on Thermal Efficiency

A comparison of the saturated, superheat, and reheat steam cycles is shown on the temperature-entropy diagram, Fig. 1.

In the saturated-steam cycle, feedwater is heated to its saturation temperature along line *AB* and evaporated along line *BC*. Steam then is admitted to the turbine and, in an ideal turbine, expands along the isentropic line *CD*. Exhaust steam is then condensed along line *DA*, returning to its initial point *A*.

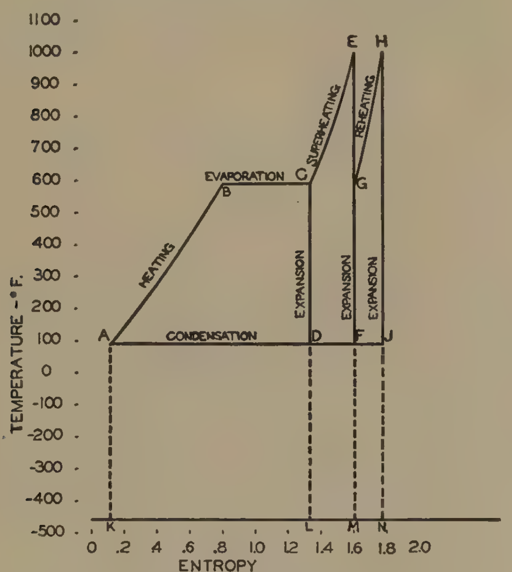

Fig. 1 Temperature-Entropy Diagram

In this cycle the total heat added is represented by area *ABCLK*, where *KL* is drawn at absolute zero, or —459.7 F. The heat rejected to the condenser circulating water is represented by area *ADLK*. Thus the heat converted into useful work, assuming an ideal turbine, becomes area *ABCD* and the thermal efficiency of the cycle the ratio of area *ABCD* to area *ABCLK*.

In the superheat cycle, feedwater is heated along line *AB* and evaporated along line *BC* as in the saturated-steam cycle. It is then superheated along *CE* before expanding through an ideal turbine along isentropic line *EF* and condensing along line *FA*. The incremental heat available for work is represented by area *CEFD* and, from inspection, it is apparent that the ratio of this added work to the heat added (area *CEML*) is greater than that of the saturated-steam cycle. This means that the superheat-cycle efficiency is greater than that of the saturated-steam cycle.

In the superheat cycle, the actual turbine efficiency increases because of the reduction of moisture in the low-pressure stages, which adds further to the thermal advantage to be obtained from the use of superheat.

In the reheat cycle, steam, after expanding through part of the turbine, such as to point *G*, is resuperheated to point *H*, after which it expands along line *HJ* to the exhaust. The additional heat added to the steam is represented by area *GHNM* and the additional heat rejected to the condenser cooling water area *FJNM*.

Therefore it is evident that the use of reheating increases available heat from each pound of steam but at the expense additional heat rejected to the condenser cooling water. I further evident that, if the ratio of the incremental heat conver into work to the incremental heat added is greater than corresponding ratio under the superheat cycle, a gain in ther efficiency, and, consequently, a reduction in fuel-consumpt rate, is accomplished.

The point on the expansion line at which this reheating ta

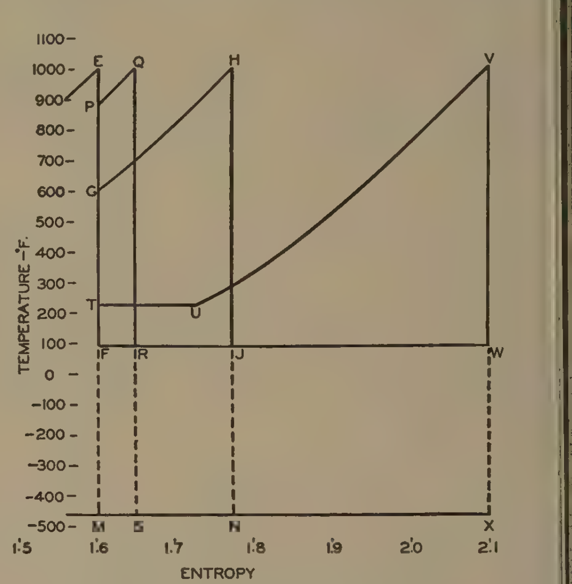

Fig. 2 Temperature-Entropy Diagram

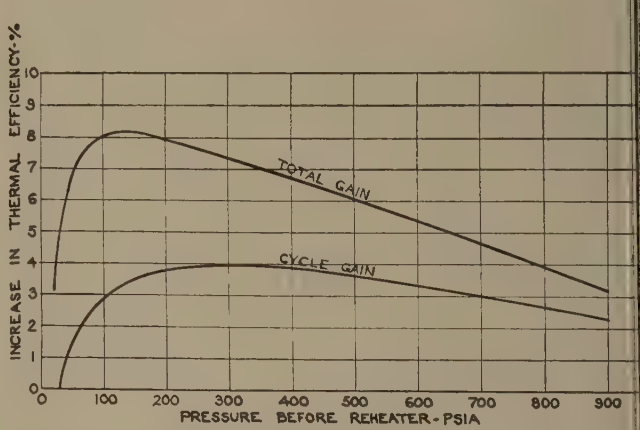

Fig. 3 Increase in Thermal Efficiency Due to Reheat
(1450 psig, 1000 F initial temperature, 1000 F reheat temperature, 1.5 in. abs; straight condensing operation, no pressure drop through reheate

place has a decided effect on the gain in thermal efficiency to realized by reheating. In order to illustrate this more clearl the reheat portion of the temperature-entropy diagram has be enlarged in Fig. 2. It will be seen that if this takes place at t upper part of the expansion, such as at point *P*, the thermal e ficiency of the reheat part of the cycle, represented by the ratio area *PQRF* to area *PQSM*, is relatively high. However, sinc this increment is only a small part of the total work done, th over-all effect on the thermal efficiency is comparatively smal If, conversely, the reheat takes place at the lower part of the e pansion, such as at point *T*, the incremental work done is a great

part of the total but its efficiency, represented by the ratio of area $TUVWF$ to area $TUVXM$, may be even lower than that of the nonreheat cycle, thus causing the reheat to have a detrimental rather than beneficial effect on the over-all thermal efficiency. It is therefore apparent that there is some intermediate optimum point where the greatest gain can be realized. This will be indicated on curves to be shown later.

The reheat cycle has the same additional advantage as the superheat cycle, in that it further reduces the moisture in the low-pressure stages, and in this way increases the efficiency in this section of the turbine.

The curves in Fig. 3 illustrate the increase in thermal efficiency, which corresponds to the reduction in heat consumption, to be realized from the use of reheating at various reheat pressures.

The following assumptions have been made:

(*a*) Steam conditions—1450 psig, 1000 F initial temperature, 1000 F reheat temperature, 1.5 in. Hg abs exhaust pressure.

(*b*) Pressure drop through reheaters and reheater piping—zero.

(*c*) No extraction of steam for regenerative feedwater heating.

The lower curve, labeled cycle gain, shows the increase in thermal efficiency obtained from the reheat cycle alone, assuming ideal turbine efficiency. The upper curve, labeled total gain, includes not only the gain from the cycle but also reflects the improvement in turbine efficiency due to the elimination of, or reduction in, the amount of moisture in the low-pressure section.

Similar curves for other steam conditions indicate that the shapes of the curve of total gain, plotted against the ratio of reheat to initial steam pressure, are practically identical for steam pressures from 600 to 2000 psig and for straight condensing operation with steam reheated to its initial temperature. It will be noted that the optimum gain from the use of reheat is about $8^1/_4$ per cent, with this maximum improvement occurring at a reheat pressure of about 10 per cent of the initial pressure.

Unfortunately, an improvement of this magnitude cannot be realized in an actual installation because of several factors, the most important of which are the pressure drop through the reheater and its piping, the number of stages of regenerative feedwater heating and, in some cases, the difference between the reheat and initial steam temperatures.

Effect of Pressure Drop Through Reheater

Zero pressure drop through the reheater and the piping to and from the reheater cannot of course be realized. In most cases, this pressure drop is of the order of 10 per cent of the absolute pressure at the exhaust of the high-pressure section of the turbine. Lower values can be obtained by the use of liberal-size reheaters, piping, and valves, but the cost of such oversize equipment usually cannot be justified economically.

The effect of pressure drop is shown in Fig. 4. This curve indicates that the percentage gain in thermal efficiency obtainable with reheat becomes only about 90 per cent of the values shown in Fig. 3 when a 10 per cent pressure drop occurs in the reheater and its piping. Thus a loss of 1 per cent of the percentage gain in thermal efficiency is suffered for each per cent pressure drop, this loss being somewhat greater for reheat pressures below and above the optimum pressure.

Effect of Regenerative Feedwater Heating

The use of the regenerative cycle, or extraction of steam for heating the turbine's own feedwater, has a compensating effect on the gain realized from the reheat cycle. By this is meant that the greater the number of stages of regenerative feedwater heating, the smaller the gain to be realized from the reheat cycle.

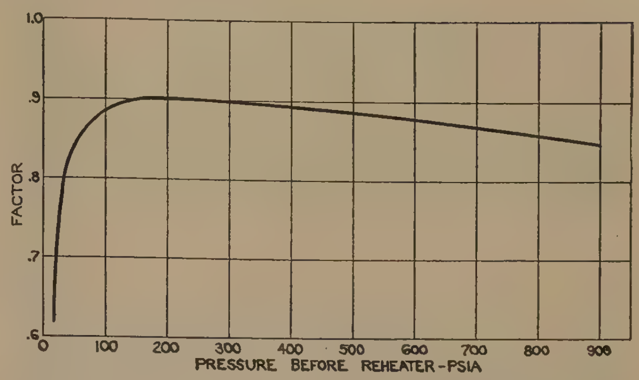

Fig. 4 Factor to Be Applied to Increase in Thermal Efficiency Due to Reheat for 10 Per Cent Pressure Drop Through Reheater

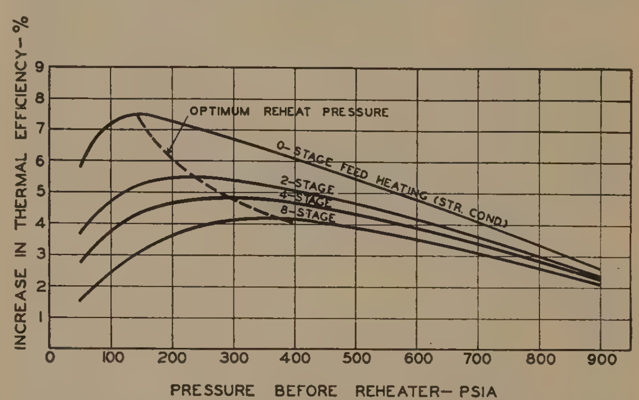

Fig. 5 Effect of Regenerative Feedwater Heating on Increase in Thermal Efficiency Due to Reheat
(1450 psig, 1000 F initial temperature, 1000 F reheat temperature, 1.5 in. Hg abs; 10 per cent pressure drop through reheater.)

The effect of regenerative feedwater heating on the gain in thermal efficiency is illustrated in the curves in Fig. 5. From these curves it will be seen that this gain decreases as the number of stages of steam extraction increases.

These curves also show that regenerative feedwater heating not only influences the gain derived from reheat but also the optimum reheat pressure. Whereas this optimum reheat pressure is about 10 per cent of the initial pressure for straight condensing operation, maximum gain in thermal efficiency will be obtained at about 20 per cent of the initial pressure for 4-stage regenerative feedwater heating operation. The gain from reheat for five stages of extraction for regenerative feedwater heating and for various steam conditions is shown on the following curves:

850 psig	900 F	Fig. 6
1250 psig	950 F	Fig. 7
1450 psig	1000 F	Fig. 8
1800 psig	1050 F	Fig. 9

On these and subsequent curves, the term "pressure before reheater" means the steam pressure at the exhaust of the high-pressure turbine before the steam enters the piping to the reheater.

These curves indicate that the gain in thermal efficiency obtainable with reheat to its initial steam temperature is about 4.6 per cent at optimum reheat pressure, this optimum pressure being slightly less than 25 per cent of the initial pressure. They further indicate that this reheat pressure can deviate materially from the optimum pressure without substantial loss. For example, an improvement in thermal efficiency of better than 4

per cent can be realized for a range in reheat pressure of from 11 to 37 per cent of the initial steam pressure.

These curves indicate values of increase in thermal efficiency for a range in reheat temperature of 100 F above and below the initial steam temperature. It will be recognized that temperatures in excess of 1050 F, shown in Figs. 8 and 9, are in excess of present central-station operating experience.

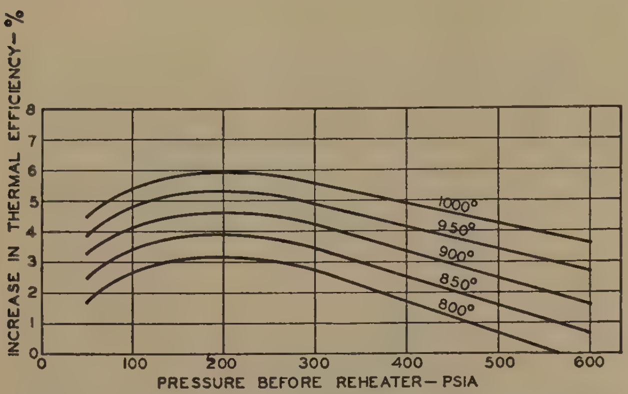

Fig. 6 Increase in Thermal Efficiency Due to Reheat
(850 psig, 900 F total temperature, 1.5 in. Hg abs; five-stage feed-heating operation, 10 per cent pressure drop through reheater.)

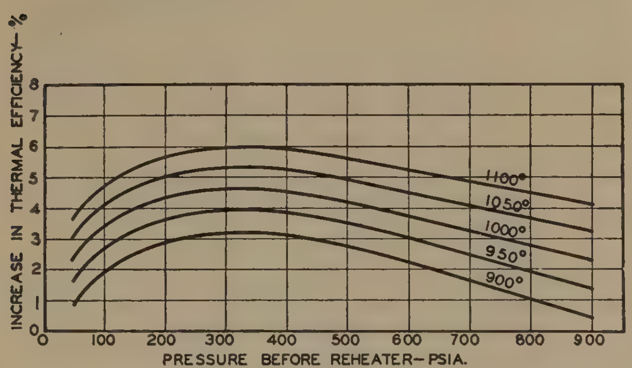

Fig. 8 Increase in Thermal Efficiency Due to Reheat
(1450 psig, 1000 F total temperature, 1.5 in. Hg abs; five-stage feed-heating operation, 10 per cent pressure drop through reheater.)

Effect of Reheat Temperature

The effect of reheat temperature on the increase in thermal efficiency is also indicated on the curves in Figs. 6, 7, 8, and 9. As might be expected, the thermal gain from reheat is increased by an increase in reheat temperature.

Other factors which may affect thermal efficiency of the reheat turbine are the type of turbine construction and, to a fairly minor extent, reduction in turbine leaving losses.

The following formula can be used, within fairly broad limitations, to estimate the increase in thermal efficiency obtainable from the use of reheat at optimum reheat pressure

$$r = 8.25\left(1 - \frac{\Delta p}{100}\right)\left(1 - \frac{\sqrt{n}}{6}\right)\left(1 - \frac{t_1 - t_r}{380}\right)$$

where

r = reduction in heat-consumption rate realized by reheat, per cent

Δp = pressure drop through reheater and reheater piping; per cent of absolute pressure before reheater

n = number of stages of regenerative feedwater heating

t_1 = initial steam temperature, deg F

t_r = reheat steam temperature, deg F

For example, if we assume steam conditions of 1450 psi 1000 F initial temperature, 1000 F reheat temperature, 1.5 in. H abs with six-stage extraction for feedwater heating, and with

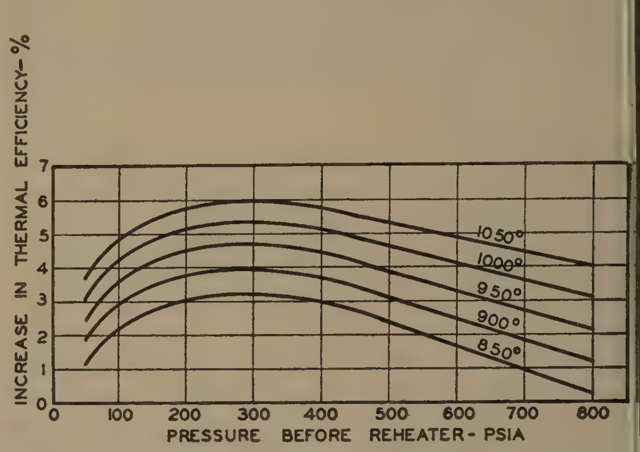

Fig. 7 Increase in Thermal Efficiency Due to Reheat
(1250 psig, 950 F total temperature, 1.5 in. Hg abs; five-stage feed-heatin operation, 10 per cent pressure drop through reheater.)

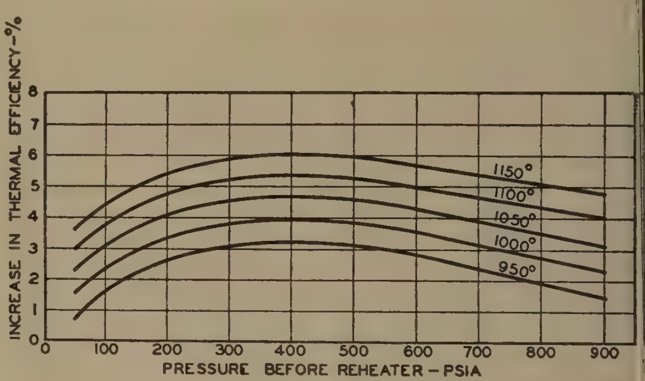

Fig. 9 Increase in Thermal Efficiency Due to Reheat
(1800 psig, 1050 F total temperature, 1.5 in. Hg abs; five-stage feed-heating operation, 10 per cent pressure drop through reheater.)

10 per cent pressure drop through the reheater, the reduction in heat rate will be approximately as follows

$$r = 8.25\left(1 - \frac{10}{100}\right)\left(1 - \frac{\sqrt{6}}{6}\right)\left(1 - \frac{1000 - 1000}{380}\right)$$

$$= 4.4 \text{ per cent}$$

Thus if the heat-consumption rate of a nonreheat turbine operating under steam conditions of 1450 psig, 1000 F, 1.5 in. Hg abs is 8700 Btu per kwhr, the corresponding heat-consumption rate for 1000 F reheat under the conditions listed would become 8700 × 0.956, or 8317 Btu per kwhr.

Effect on Steam Flow

The steam flow to the turbine is reduced materially by the use of reheat. This is shown in Fig. 10 for initial steam conditions of 1450 psig, 1000 F. Reductions in steam flow of about this same order are realized for other operating steam conditions.

This reduction in steam flow will reduce correspondingly the size of the boiler and high-pressure steam piping and valves.

The size of feedwater heating and boiler feed-pump equipment is also reduced correspondingly.

Effect on Exhaust Flow

The exhaust-steam flow will be reduced by the use of reheat, as shown in Fig. 11 for steam conditions of 1450 psig, 1000 F. However, the reduction in the volume of exhaust steam is somewhat less because of its greater specific volume, as will be seen from the curve in Fig. 12.

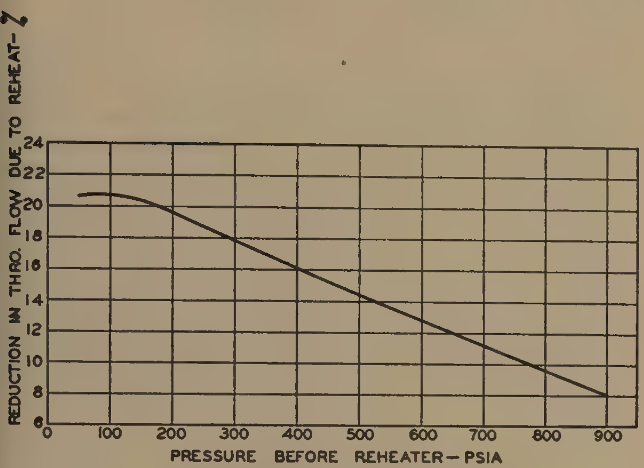

Fig. 10 Reduction in Steam Flow to Throttle Due to Reheat
(1450 psig, 1000 F initial temperature, 1000 F reheat temperature, 1.5 in. Hg abs; five-stage feed-heating operation, 10 per cent pressure drop through reheater.)

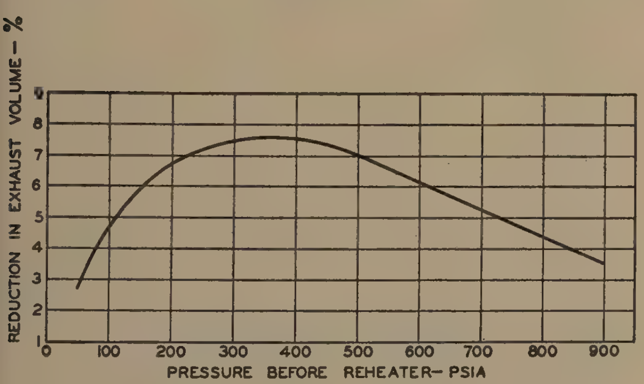

Fig. 12 Reduction in Exhaust Volume Due to Reheat
(1450 psig, 1000 F initial temperature, 1000 F reheat temperature, 1.5 in. Hg abs; five-stage feed-heating operation, 10 per cent pressure drop through reheater.)

The heat rejected to the condenser is reduced to a lesser degree than the exhaust-steam flow, since the enthalpy of the exhaust steam is greater with the reheat than with the nonreheat cycle. This reduction in the heat rejected to the condenser cooling water is shown in Fig. 13. This reduction in the heat transfer in the condenser will reduce correspondingly the size of the condenser and its auxiliary equipment.

Comparison of curves in Figs. 10, 11, and 13 discloses that the greatest reduction in throttle- and exhaust-steam flows is obtained at a rather low reheat steam pressure, about 7 per cent of the initial pressure, whereas the greatest reduction in heat absorbed by the condenser occurs at a much higher reheat steam pressure, about 25 per cent of the initial pressure. This is explained by the fact that the lower the reheat steam pressure the greater the amount of heat added in the reheater and the heat available for useful work with, however, an increase of the heat rejected to the condenser cooling water.

Effect on Exhaust Moisture

As stated previously, reheat reduces materially the amount of moisture passing through the low-pressure section of the turbine. The effect on exhaust moisture for steam conditions of 1450 psig, 1000 F initial temperature, 1000 F reheat temperature, 1.5 in. Hg abs exhaust pressure is illustrated by the curves in Fig. 14.

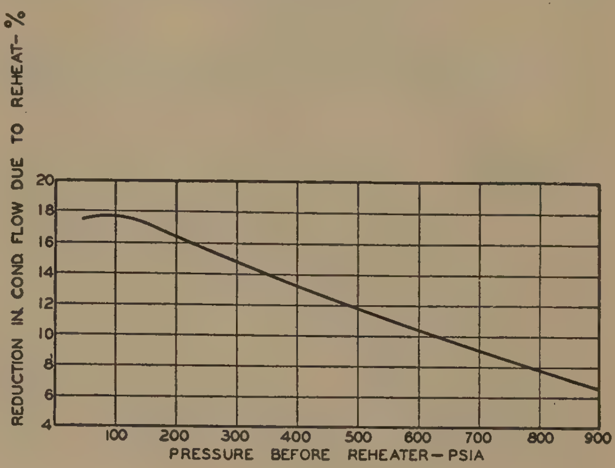

Fig. 11 Reduction in Steam Flow to Condenser Due to Reheat
(1450 psig, 1000 F initial temperature, 1000 F reheat temperature, 1.5 in. Hg abs; five-stage feed-heating operation, 10 per cent pressure drop through reheater.)

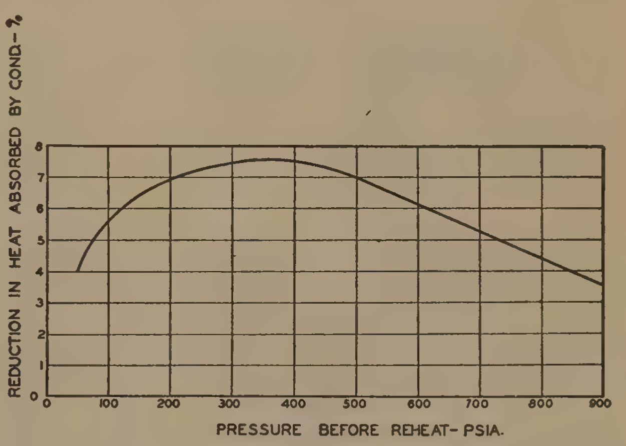

Fig. 13 Reduction in Heat Absorbed by Condenser Due to Reheat
(1450 psig, 1000 F initial temperature, 1000 F reheat temperature, 1.5 in. Hg abs; five-stage feed-heating operation, 10 per cent pressure drop through reheater.)

Turbine Design

Three types of turbine construction are now being considered by turbine builders. These three types are shown diagrammatically in Fig. 15.

In arrangement (A) three cylinders are used. Steam, after expanding through the high-pressure element, is reheated, after which it expands through the intermediate and low-pressure elements in series.

In arrangement (B) high-pressure steam enters near the center of the combined high and intermediate-pressure element, passing in one direction to the one end. After being reheated, the steam again enters near the center and passes in the opposite

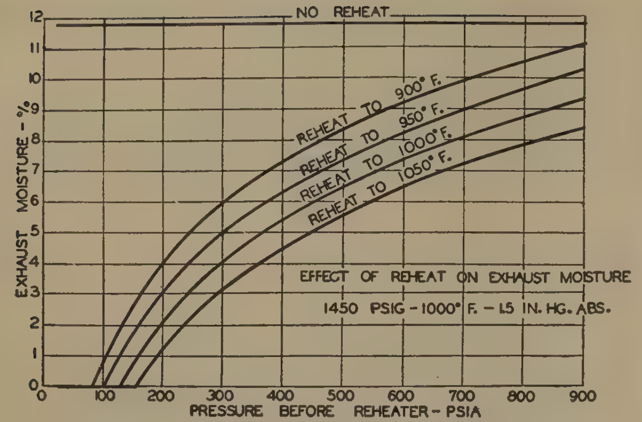

Fig. 14 Effect of Reheat on Exhaust Moisture
(1450 psig, 1000 F, 1.5 in. Hg abs.)

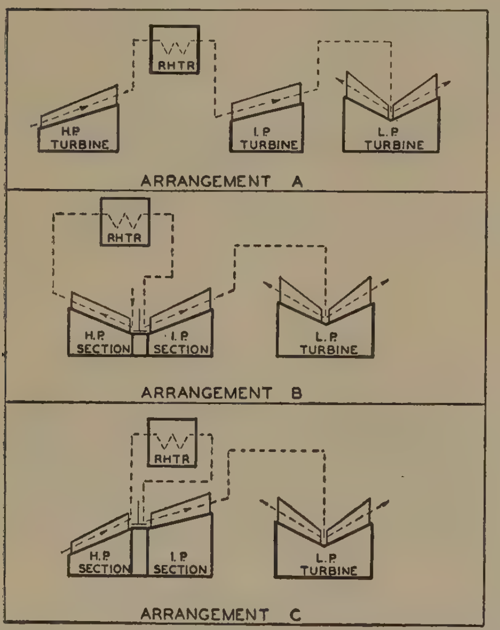

Fig. 15 Diagrammatic Arrangement of Types of Turbine Construction

direction to the other end of the same turbine element before passing to the low-pressure element.

In arrangement (C) high-pressure steam enters one end, exhausts to the reheater near the center, after which the steam again enters near the center and passes in the same direction to the low-pressure end, before passing to the low-pressure element.

Advantages and disadvantages can be offered for each of the arrangements illustrated. Arrangement (A) is slightly more efficient, is more conservative in that the high- and intermediate-pressure elements are separated into two cylinders, and thus involve shorter individual elements. It does, however, result in greater unit length, a factor which must be balanced against the benefits of slightly greater efficiency and more conservative design.

Arrangement (B) has the advantage of keeping the higher temperature of the steam from both the main and reheat boiler in a relatively short section near the center of the high-pressure element. However, because of the pressure drop across the diaphragm, the leakage of steam through this diaphragm results in some loss in efficiency and also in some control difficulties.

Arrangement (C) has greater temperature difference across the center diaphragm, but has the advantage of having practically no pressure drop across this diaphragm.

Control Problems

The design of reheat turbine units introduces novel problems in the control of steam to provide adequate protection to the turbine under all possible normal and abnormal conditions of service. For instance, in case of sudden and complete loss of load, not only must high-pressure steam flow be cut off quickly but, in addition, the steam entrained in the reheater and its piping must be prevented from returning to the turbine. Other considerations enter into the operation of starting and putting such turbines on the line, and of absorbing rapid load changes. In these matters, close co-operation between designers of the plant, boiler, turbine, and auxiliary equipment is essential to success.

Conclusion

In this paper we have shown the advantages and disadvantages of the reheat cycle. The advantages are as follows:

(a) Material reduction in heat and fuel-consumption rates.

(b) Reduction in size and cost of main boiler, condenser, and heater equipment.

(c) Reduction in exhaust moisture, resulting in lower blade erosion and higher stage efficiency in the low-pressure section.

The disadvantages include the following:

(a) Greater unit length.

(b) Higher costs of turbine and station piping.

(c) Additional cost of reheater equipment.

In weighing the advantages against the disadvantages of the reheat cycle, it becomes apparent that the future adoption of this cycle will depend to a great extent upon the cost of fuel and upon the load factor of the unit. At the present time, this cycle often will prove economically desirable on larger units where fuel costs and load factors are high.

General operating experience has been favorable, resulting in low station heat-consumption rates with a high degree of reliability and no severe operating problems.

If fuel costs remain constant or decrease and if, in addition, metallurgical developments produce materials permitting substantially higher steam temperatures without sacrifice in reliability and long life, the reheat cycle will again fall into disuse as it did several years ago. If, on the other hand, fuel costs continue to increase, as now appears probable, this cycle can be justified for smaller and smaller units and will become increasingly popular.

Acknowledgments

The author wishes to acknowledge the aid of Mr. C. B. Campbell for his guidance and constructive criticism and of Mr. H. R. Reese for the preparation of curves and other data appearing in this paper.

Steam-Generating Equipment for Resuperheating Cycles

By MARTIN FRISCH,[1] NEW YORK, N. Y

This paper examines some of the technical and operating problems which must be faced in the selection of equipment to supply steam to high-pressure turbine-generating equipment employing steam resuperheated to high temperatures in the low-pressure section. The costs of steam-generating equipment per kilowatt of turbine-generator capability for resuperheating cycles are compared with costs of equipment for cycles which do not employ resuperheating. The effects on equipment costs, of furnace-exit temperature limitations imposed by fuel quality and load range over which primary and resuperheated steam temperatures must remain constant, are examined in detail. Data are presented comparing present-day prices of steam-generating equipment per kilowatt of turbine-generator capability for several high-temperature and high-pressure resuperheating and nonresuperheating cycles usually employed in power stations.

SEVERAL new central-station units employing high-temperature resuperheating cycles will soon be operating. These units, designed to conserve fuel, were planned by farsighted utilities operators to offset rising fuel costs.

For many years one prominent utility in the Middle West has operated several units with turbine conditions as follows:

Primary throttle steam at 1300 psig and 830 F; resuperheated steam at 375 psig and 835 F.

The two resuperheating cycles recently given most serious consideration involve turbine conditions as follows:

Cycle Br:

Primary steam to high-pressure turbine at 1450 psig and 1000 F; resuperheated steam to low-pressure turbine at 400 psig and 1000 F.

Cycle Ar:

Primary steam to high-pressure turbine at 2000 psig and 1050 F; resuperheated steam to low-pressure turbine at 400 psig and 1000 F.

The design of steam generators for these conditions poses certain problems which do not have to be considered when resuperheaters are not included.

The purpose of this paper is to review these problems and to determine how their necessary solutions affect the design and the cost of steam generators per kilowatt of turbine-generator capability for resuperheating units, as compared with nonresuperheating units. Two types of units will be considered: (1) Units utilizing convection surface only for superheating and resuperheating; and (2) units utilizing radiant superheaters in combination with convection surface.

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Superheating and Resuperheating Only by Convection

Resuperheating units utilizing convection surface only for superheating and resuperheating are more difficult to design for equal fuel flexibility and reliability than nonresuperheating units for the same primary steam conditions. In order to illustrate this, the design problems posed by the now most seriously considered resuperheating cycles Ar and Br are compared with those of the corresponding nonresuperheating cycles An and Bn for the same primary steam conditions.

In Fig. 1 the operating conditions of these cycles are tabulated and curves are shown of the probable turbine-generator heat rates and the corresponding primary steam-throttle flow quantities for steam turbine-generator units of various capabilities from 50,000 to 150,000 kw. The operating conditions, probable heat, and throttle-flow quantities of one other resuperheating and three other nonresuperheating cycles usually employed are included, so that plant-cost comparisons to follow may be more readily understood.

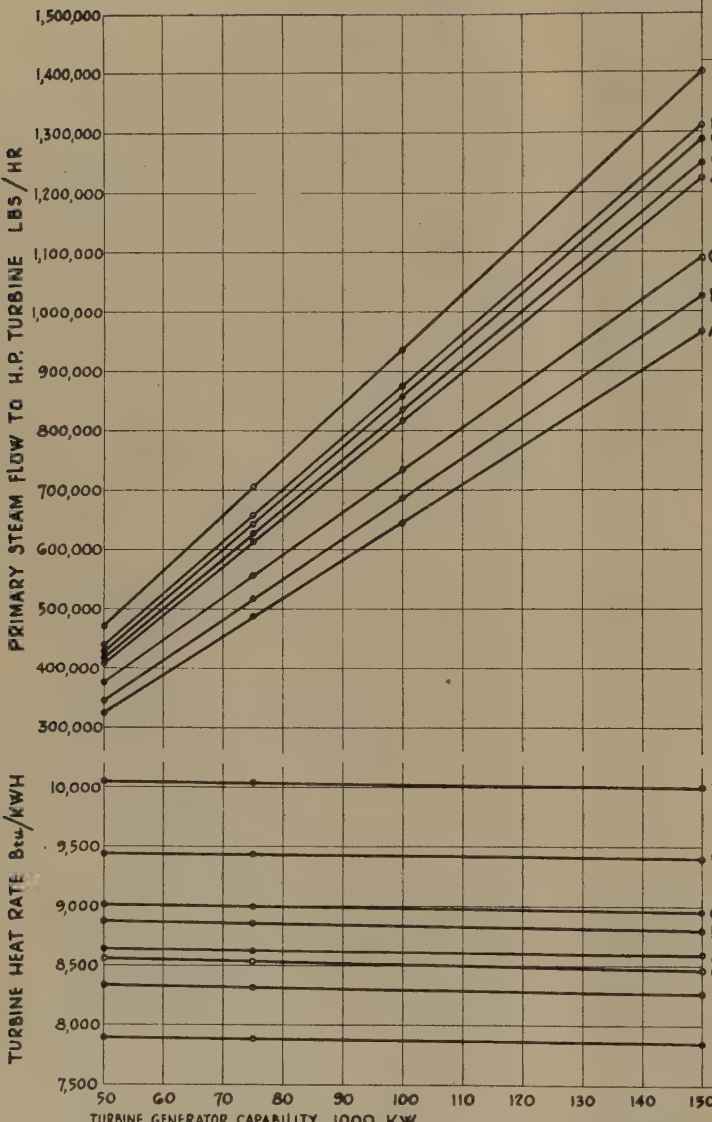
All of the cycles compared employ five stages of regenerative feedwater heating and exhaust to the condenser at 1.5-in. Hg abs. The steam generators are pulverized-coal-fired and have a full-load efficiency of 87 per cent.

A serious problem in the design of conventional arrangements of high-pressure high-temperature steam generators for resuperheating cycles, such as cycles Ar and Br, for which full primary and resuperheated-steam temperature must be realized over a wide load range, is the difficulty of providing sufficient heat in the gas leaving the furnace for this duty, unless small furnaces with higher furnace-exit temperature than had been considered desirable in the past for nonresuperheating cycles are used.

Fuel characteristics, particularly the slagging properties of the ash in the fuel will always govern furnace designs whether resuperheating is required or not. Therefore the true temperature of the furnace gas entering the superheating and resuperheating zone should be limited at every load to preclude slagging. The maximum tolerable temperature of gases flowing into closely spaced convection heat-absorbing surfaces is always lower than most purchasers think they can afford until mounting costs and labor problems, arising from necessary frequent manual deslagging, commence to bedevil them.

While some comfort may be taken from the fact that ash containing iron compounds in certain forms tends to fuse at higher temperatures in oxidizing atmospheres than in reducing atmospheres in laboratory muffles, ash particles, unfortunately, do not behave equally well in actual furnaces. Generally, each floating particle still contains unconsumed incandescent carbon as it approaches the furnace exit. As long as its own temperature and that of the film of gas adhering to it and surrounding it remain high, this incandescent carbon continues to consume oxygen left in the film or which enters it by diffusion. It then goes after the CO_2 and H_2O present and as long as sufficient heat flows into the particles from the surrounding gas to maintain the endothermic reducing process, the particles will be in a reducing film. So it is most unlikely that the gas film which surrounds the particle will provide that much-to-be-desired oxidizing atmosphere.

Experience teaches that the accumulation of sticky particles on



	PRESSURE								TEMP. °F.		
	Psig							in.Hg			
CYCLE	BOILER DESIGN	SUPRHTR OUTLET	H.P. TURBINE INLET	H.P. TURBINE EXHAUST	RESUPRHTR INLET	RESUPRHTR OUTLET	L.P. TURBINE INLET	L.P. TURBINE EXHAUST	FEED WATER 5 STAGE REGEN. HEAT'G.	PRIMARY STEAM AT TURBINE	RESUPRHTR STEAM AT TURBINE
En	750	675	600	—	—	—	—	1.5	376	825	—
Dn	1025	925	850	—	—	—	—	1.5	402	900	—
Cn	1500	1325	1250	—	—	—	—	1.5	450	950	—
Bn	1700	1500	1450	—	—	—	—	1.5	450	1000	—
An	2300	2050	2000	—	—	—	—	1.5	450	1050	—
Cr	1500	1325	1250	400	390	370	360	1.5	450	950	950
Br	1700	1500	1450	425	415	395	385	1.5	450	1000	1000
Ar	2300	2050	2000	425	415	395	385	1.5	450	1050	1000

En	750	675	600	—	—	—	—	1.5	376	825	—
Dn	1025	925	850	—	—	—	—	1.5	402	900	—
Cn	1500	1325	1250	—	—	—	—	1.5	450	950	—
Bn	1700	1500	1450	—	—	—	—	1.5	450	1000	—
An	2300	2050	2000	—	—	—	—	1.5	450	1050	—
Cr	1500	1325	1250	400	390	370	360	1.5	450	950	950
Br	1700	1500	1450	425	415	395	385	1.5	450	1000	1000
Ar	2300	2050	2000	425	415	395	385	1.5	450	1050	1000

Fig. 1 Design Conditions and Steam Requirements of Typical Central-Station Steam Cycles

slag screen and superheater tubes proceeds even in very large furnaces designed to reduce furnace-gas-exit temperatures below ash-fusion temperatures. This often happens when the fineness is low and suspended particles which continue to burn smash against convection surfaces as they pass out of the furnace before freezing completely. In view of this and unavoidable temperature stratification, it is safer to predicate furnace-exit temperature on the fusion characteristics of ash in reducing atmospheres rather than in oxidizing ones.

The convection surfaces of units designed to burn only liquid fuels also sometimes slag up. In fact, with some types of oils, such units may be more troublesome than units burning coal. The correction for this must be twofold: (1) The furnace-exit temperature should be limited as for units burning coal; (2) the tubes in the convection sections should be widely spaced. These two effects sometimes combine to increase the size of the convection section to impractical dimensions and to increase the cost beyond what had been considered reasonable in the past.

The use of some radiant superheating surface makes it possible to design such oil-fired units with slagging oils at reasonable cost, since the amount of widely spaced convection surface required in such units will be less than in practical units of the convection type.

Effect of Gas Temperature and Superheat Control Range

Fig. 2 shows the effect under ideal conditions of superheat control range and gas temperature on the amount of superheater surface required for the resuperheating (Ar, Br) and nonresuper-

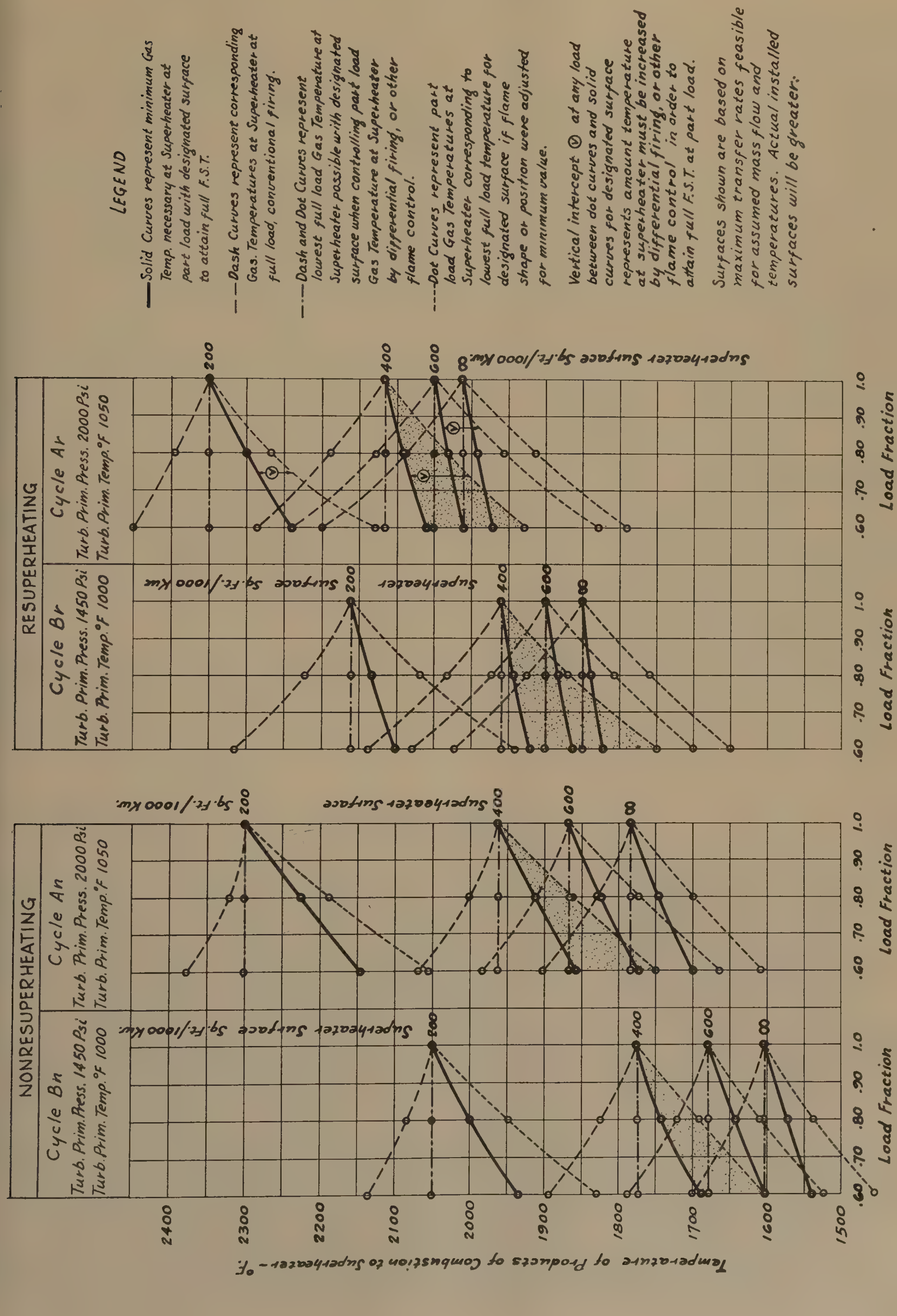


Fig. 2 Effect of Gas-Temperature and Superheat Control Load Range on Superheater Size

heating (An, Bn) cycles whose characteristics are summarized in Fig. 1, when only convection superheaters and resuperheaters are used. Fig. 2 shows that as the primary-steam pressure and temperature increase, the minimum gas temperature required to produce a prescribed final steam temperature at any load with a given amount of superheating surface increases substantially. There is a further increase if resuperheaters are added. The full import of this may be appreciated by examining the solid curves in Fig. 2 which define the minimum gas temperature necessary to attain full final steam temperature at every fraction of full load from 60 to 100 per cent with constant amounts of superheating surface varying from 200 to infinity sq ft per 1000 kw for each of the cycles Ar, Br, An, and Bn. The surface values shown are the smallest that are ideally possible with perfect cleanliness, perfect gas and temperature distribution, reasonable draft loss, and safe metal temperatures.

Superheater selections for actual installations would be larger. Superheaters larger than about 400 sq ft per 1000 kw approach the impractical in size. With 400 sq ft of superheating surface per 1000 kw under ideal conditions the full-load gas temperature to the superheater will be no less than 1960 F for the 1450-psig 1000-deg F resuperheating cycle (Br). This corresponds to a furnace-exit temperature of about 2030 to 2050 F. This full-load gas temperature is sufficient only if the full final steam temperature is not required below top load, or if the pronounced tendency of the furnace-exit-gas temperature to fall as the load is reduced (as revealed by the dot curves) is offset by differential firing. If full final temperature is required at fractional loads as low as 70 per cent, the full-load gas temperature at the superheater, corresponding to the minimum gas temperature of 1935 deg at the superheater required at 70 per cent of full load, will be 2080 F. This corresponds to furnace-exit temperature of 2150 F.

Correspondingly for the 2000-psig 1050-deg F resuperheating cycles, the minimum full-load temperature at the superheater with 400 sq ft per 1000 kw would be 2115 deg with a corresponding furnace-exit temperature of about 2185 F. For control down to 70 per cent of top load, where a minimum temperature of 2075 F is required at the superheater, the corresponding temperature at the superheater at full load would be 2235 F and the furnace-temperature 2305 F, unless the surfaces are selected on the bas differential firing. With the same amount of surface per kilo the corresponding nonresuperheating cycles require temp tures approximately 200 deg F less for equal control range.

The reason for this is readily apparent from the following siderations:

The heat available at any load for superheating and resu heating is determined by the weight of gas, its temperature en ing the superheating and resuperheating zone and the lowest t perature to which the gas may be cooled before it escapes from superheater surface. This terminal difference seldom may be duced to less than about 100 deg F at the minimum superheat- trol load without making the surface impractically large.

The weight of gas at any load is determined by the efficie and excess air. Its temperature is determined by the extent effectiveness of the furnace heat-absorbing surfaces, excess flame volume, shape, location, and distribution with respect to furnace heat-absorbing surfaces. Some engineers overlook importance of some of these factors and use only the heat lib tion in the furnace per square foot of projected furnace absorb surface as the basis for passing judgment on the adequacy furnace and the furnace-exit temperature to be expected. Chan in these factors may affect the furnace-exit temperature hundreds of degrees at the same liberation rate.

Table 1 sets forth the pertinent conditions which determ part-load superheater and resuperheater duty for the high-p sure resuperheating and nonresuperheating cycles under disc sion, together with theoretical minimum temperatures of gas the superheater, and the estimated corresponding maxim furnace-exit temperature already referred to in the discussion Fig. 2.

Steam-generator prices are sensitive to arrangement and p portions of the components. Therefore it is well to consider effects of steam conditions, furnace-exit temperature, and stea temperature control on surface proportioning in steam-genera design.

Fig. 3 illustrates the effect of these factors and makes it possi

TABLE 1 STEAM-GENERATOR DESIGN CONDITIONS FOR RESUPERHEATING AND NONRESUPERHEATING CYCLES

CYCLE			NON RESUPERHEATING						RESUPERHEATING					
			Bn			An			Br			Ar		
PRIMARY STEAM	Pressure at H.P. Turbine Throttle	psig	1450			2000			1450			2000		
	Percent of Full Load Throttle Flow	%	60	80	100	60	80	100	60	80	100	60	80	10
	Pressure at Superheater Outlet	psig	1455	1467	1485	2005	2017	2035	1455	1467	1485	2005	2017	203
	Pressure at H.P. Turbine Exhaust	psig	—	—	—	—	—	—	260	345	425	260	345	42
	Temperature at H.P. Turbine Throttle	°F.	1000	1000	1000	1050	1050	1050	1000	1000	1000	1050	1050	105
	Temperature at H.P. Turbine Exhaust	°F.	—	—	—	—	—	—	620	660	700	620	660	70
	Temperature of Feed Water	°F.	400	425	450	400	425	450	400	425	450	400	425	45
	Heat to Primary Steam per Lb. Pri. St.	Btu.	1117	1090	1062	1131	1104	1076	1117	1090	1062	1131	1104	107
	Lb. Gas per Lb. Primary Steam (87 % Efficiency — 20 % Excess Air)	Lb.	1.235	1.203	1.173	1.250	1.220	1.19	1.434	1.386	1.333	1.453	1.40	1.3
RESUP'RHTD. STEAM	Pressure at L.P. Turbine Inlet	psig	—	—	—	—	—	—	245	316	385	245	316	38
	Temperature at L.P. Turbine Inlet	°F	—	—	—	—	—	—	1000	1000	1000	1000	1000	100
DUTY	Heat Absorbed in Unit per Lb. Pri. Steam	Btu.	1117	1090	1062	1131	1104	1076	1296	1252	1207	1310	1266	122
	Heat Absorbed in Superheater Pri. Steam	Btu.	326	327	330	377	382	387	326	327	330	377	382	38
	Heat Absorbed in Superheater and Resup'rht'r per Lb. Primary Steam	Btu.	325	327	330	377	382	387	504	489	475	556	544	53
	per Lb. Gas	Btu.	264	272	281	302	313	325	351	353	356	383	388	39
GAS TEMP. FOR REQUIRED DUTY	Theoretical Minimum (for 0 °F. Terminal Diff. at Superh'tr. Exit & Infinite Superh'tr Surface)	°F.	1545	1570	1605	1705	1745	1785	1825	1840	1850	1965	1990	201
	Theoretical Minimum (for Terminal Difference at Superh'tr Outlet 100 °F.)	°F.	1630	1660	1695	1810	1835	1845	1915	1925	1935	2060	2080	210
	Corresponding Normal Full Load Furnace Exit Temp. (Standard Firing)	°F.	1860	1790	1755	2070	1985	1895	2190	2070	1985	2330	2220	215
	Corresponding Minimum Full Load Furnace Exit Temp. Possible With Differential Firing	°F.	1755	1755	1755	1895	1895	1895	1985	1985	1985	2155	2155	215

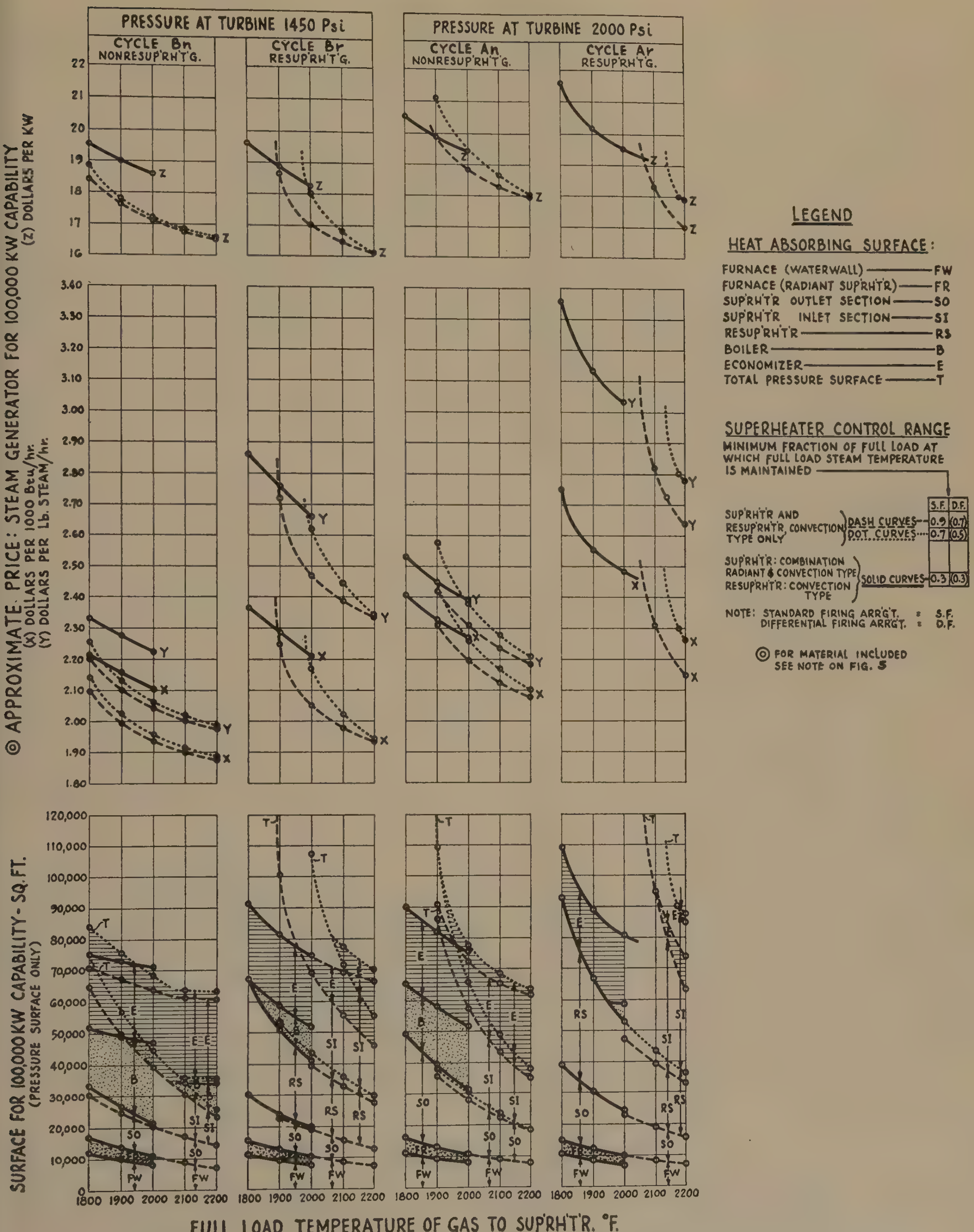


Fig. 3 Effect of Gas-Temperature and Superheat Control Range on Steam-Generator Surface Make-Up, and Price for 100,000-Kw Turbine-Generator Capability

to sense the complicated effects of changes in superheat-control range, and in the temperature of the products of combustion available to the superheater on the make-up of the pressure surface of steam generators. The distribution and total pressure surface requirements shown by the bottom curves in Fig. 3 are for steam generators which might be designed for several 100,000 kw units, each working in accordance with one of the cycles designated. To make the surface comparisons valid, the transfer rates and surface arrangements were selected for the duty so as to produce with safe metal temperatures substantially the same draft loss and gas temperature leaving the pressure surface to enter the air heater at 700 F at full load.

It should be of interest that in the nonresuperheating units for cycles, Bn and An, the effect of gas temperature and control range on the total pressure surface required is considerably less than the effect on the amount of superheater surface necessary. In the resuperheating cycles, the effects of gas temperature and control range on the total surface are more marked. Correspondingly, as may be seen from the upper part of the figure, the price per kilowatt responds to changes in temperature and control range in a manner not to be ignored. The price data apply to steam generators whose maximum capacity is just equal to the turbine-generator requirements at maximum capability and are approximate in the absolute sense, but accurate as to the proportionate effects of design limitations.

It is quite clear that arrangements of resuperheating units with superheaters and resuperheaters only of the convection type become prohibitive when a wide superheat-control range with a low furnace-exit temperature is required. Even with perfect fire control, as may be seen from Table 1, the minimum practicable furnace-exit temperature with the 1450-psig resuperheating cycle is 1985 F, and with the 2000-psig cycle, 2155 F. These limitations preclude the choice of such steam generators for resuperheating-cycle installations in the Middle West or other regions where only coals with slagging ash are economically available.

Resuperheating Units With Radiant Superheaters

There are, and always will be, plants which must be ready to burn fuels with ash of very low softening temperatures. In such plants furnace release rates are usually limited to values which, by experience, are known to result in slag-free operation with the worst available fuels. The 1450 (Br) or 2000-psig (Ar) resuperheating cycles described cannot have the same fuel flexibility, freedom from slagging, and operating continuity as the corresponding nonresuperheating cycles, (Bn and An), unless enough radiant superheating surface is used to accomplish the required superheating duty over the required load range with lower furnace-exit temperatures than those shown to be required if only convection superheaters are used.

Several of the resuperheating installations now under construction will employ superheaters and resuperheaters only of the convection type. These units, necessarily, will have to opera with temperatures entering the superheater several hundred d grees higher than would be required or considered acceptable i conventional nonresuperheating units for the same primary-stea conditions purchased contemporaneously. It will be interesting see to what extent the choice of fuels for these units will have to restricted to nonslagging types after they go into service.

Complete freedom in the selection of fuels for resup heating units may be obtained by using superheaters and r superheaters which include radiant heat-absorbing sections. resuperheating unit of this type is now under construction for utility which has installed and operated four such units, and th year purchased a fifth. This utility, since about 1931, when adopted resuperheating, has had extremely satisfactory experien with designs including radiant superheaters and radiant resupe heaters. The design conditions of these units are given in Table

These units were designed to operate with furnace heat releas of the order of 15,000 Btu per cu ft per hr and therefore wi furnace-exit temperatures so low as to preclude slagging of th convection heating surfaces with any fuel. It would have bee impossible to design these units without the use of radiant stean superheating surfaces.

Since the use of radiant superheating and resuperheating su faces makes possible the design of steam generators which will n slag with any fuel, why do many purchasers still prefer resupe heating units, including superheaters and resuperheaters only the convection type, which are more likely to slag up except wit very good fuels? This is perhaps so because not all enginee realize that there are in the United States many units operating a pressures of 900 to 1500 psi with steam temperatures at 925 1000 F, which include radiant superheaters and resuperheater and that, in the aggregate, in these modern units difficulti attributable to the use of radiant superheaters are negligibl Since there is so much to gain in freedom from slagging and i operating reliability from the use of radiant surface for super heating and resuperheating, engineers who contemplate the in stallation of resuperheating units will find it advantageous to com pare the advantages and limitations of radiant convection com binations with self-compensating load-temperature characteris tics, as against exclusively convection arrangements with stee load-temperature curves.

Comparative Price of Resuperheating and Nonresuper heating Steam Generators

Here of course economics rears its ugly head and therefore i will be of interest to examine the effect of design and arrangemen of resuperheating and nonresuperheating units for the sam primary conditions on price. Fig. 3 shows how make-up and pric of a steam generator for a turbine generator with a capabilit of 100,000 kw are affected by design limitations. It should be o interest that resuperheating units employing radiant superheater

TABLE 2 DESIGN CHARACTERISTICS OF SOME RESUPERHEATING INSTALLATIONS WITH RADIANT SUPERHEATERS AND RADIANT RESUPERHEATERS

			SUPERHEATER					RESUPERHEATER			
No. units	Year	Kilo-watts	Flow Lb/hr	Pres-sure Psig	Temp. °F.	Surface Radiant Proj.	Sq. Ft. Con-vection	Flow Lb/hr	Pres-sure Psig	Temp. °F.	Surface Ft² Radiant only Proj.
1	1931	80,000	635,000	1,340	830	1,534	8,500	590,000	375	835	1,430
1	1942	80,000	635,000	1,340	830	1,534	8,500	590,000	375	835	1,430
1	1945	80,000	635,000	1,340	830	1,534	8,500	590,000	375	835	1,430
1	1947	80,000	635,000	1,340	920	1,600	10,950	590,000	375	884	1,680
1	1948	85,845	575,000	1,489	950	1,600	14,550	494,070	371	950	1,460

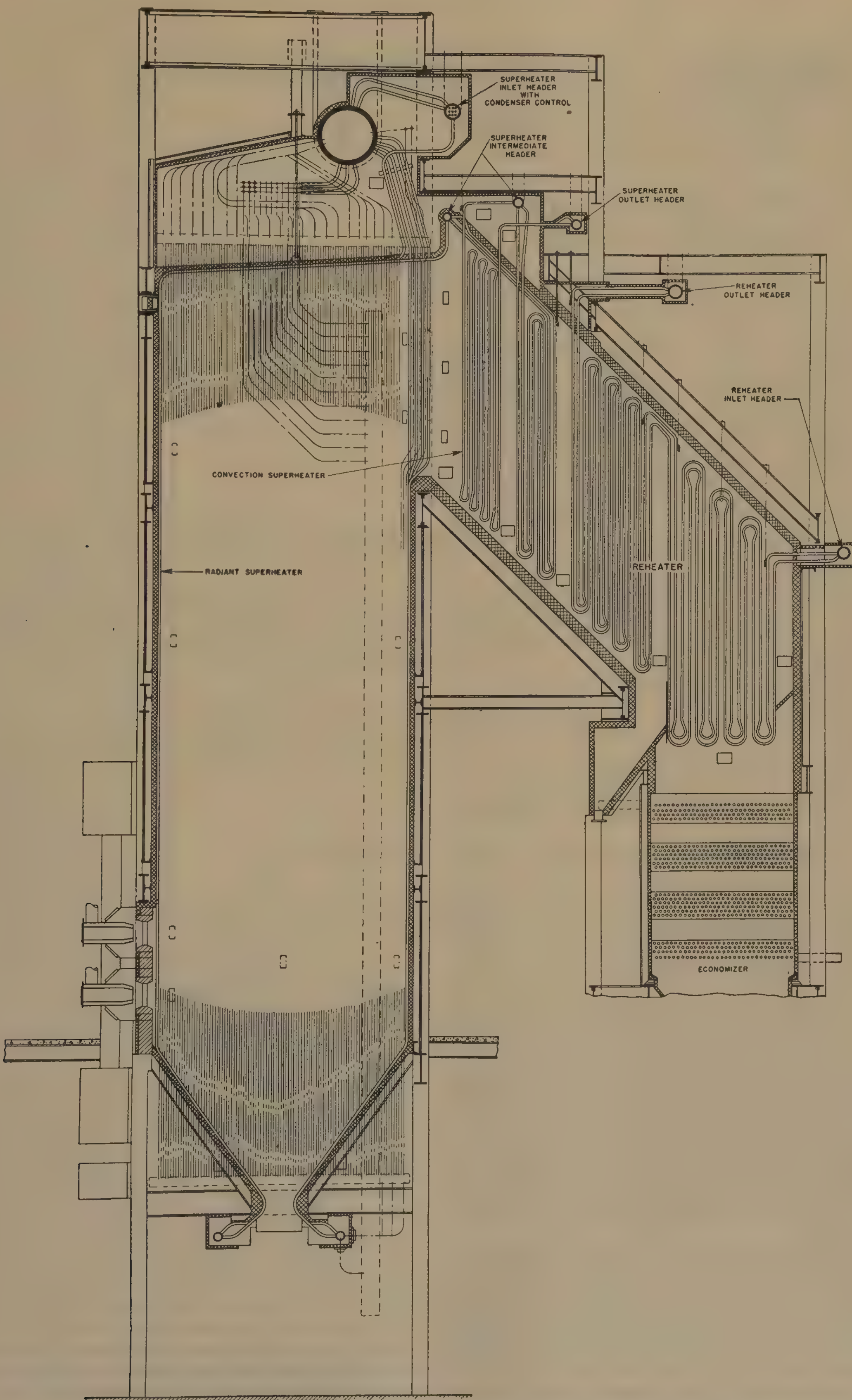


Fig. 4 Typical Resuperheating Steam Generator With Radiant Superheater

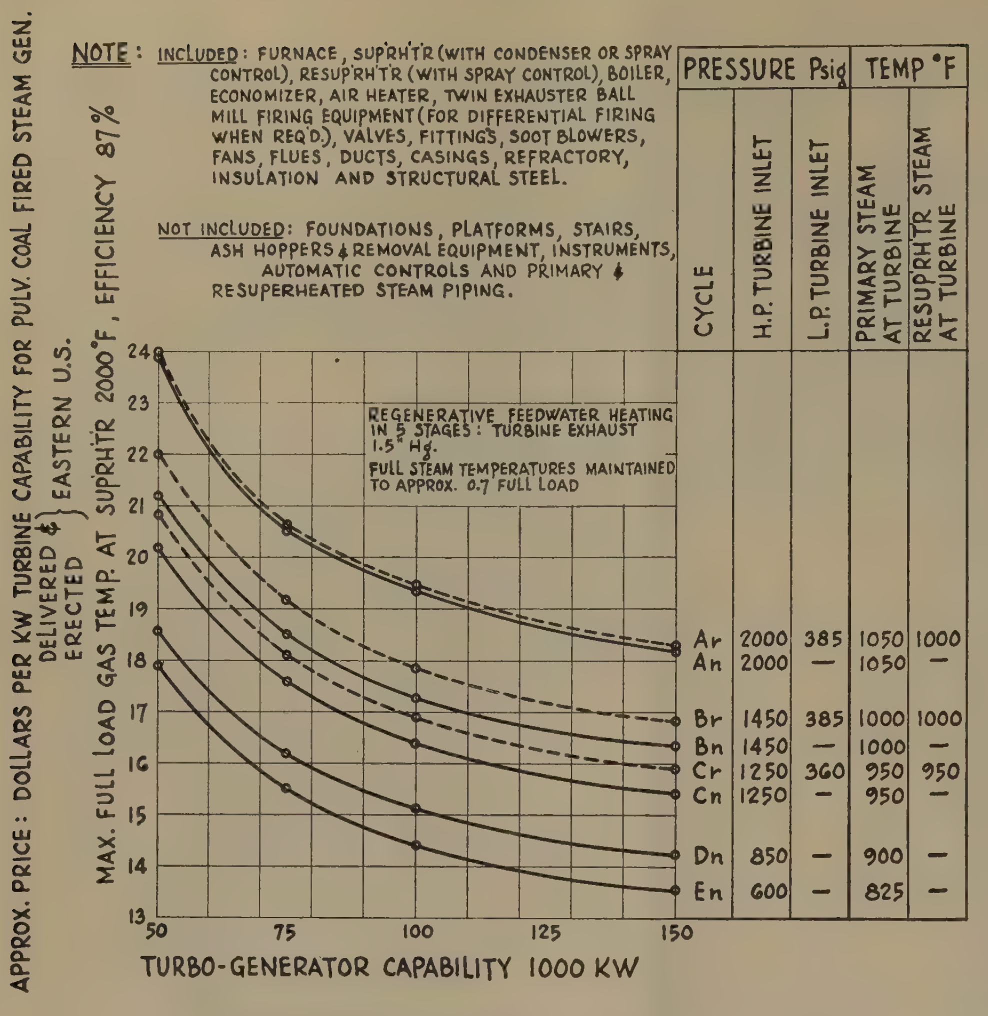


Fig. 5 Effect of Cycle and Turbine Size on Price of Steam Generators

are cheaper when fuel conditions prescribe low furnace-exit temperatures, and when operating requirements call for superheat control over a wide load range. A modern design of this type is shown in Fig. 4.

The cost of resuperheating steam generators per kilowatt turbine-generator capability is nearly the same as, and under some conditions, less than for nonresuperheating units for turbine generators of the same capability. Fig. 5 illustrates the effect of unit size and cycle type on the cost of resuperheating and nonresuperheating steam generators for a full-load gas temperature of 2000 F at the superheater, other conditions as given. The top curves in Fig. 3 show that for fuels requiring furnace-exit temperatures of less than 2000 F, resuperheating units of the convection type quickly tend to become impractical, prohibitive in price, and even impossible, whereas units with radiant sections are still economically feasible.

This disadvantage of the convection type may be offset to some extent by providing differential firing equipment which may be used to increase the furnace-exit temperature as the load is reduced. If this is done, the amount of superheating and resuperheating surface is reduced, being determined by top or near-top-load requirements rather than by part-load conditions.

Fig. 6 shows a twin-classifier twin-exhauster ball-mill arrangement which has been used for superheat control by differential firing for some 10 years, in single as well as in twin-furnace units. In single-furnace units, two rows of burners at different elevations are fired at different rates varying the relative proporti of fuel delivered to each r Each row is fired by a se rate exhauster whose out may be adjusted as requir The furnace-exit temperat may be changed by 100 or m deg by shifting the fuel prop tions between upper and lo burners. Fig. 7 shows a sin furnace resuperheating u with differential firing equ ment arranged for some f nace-exit-temperature contr

Control Characteristics Resuperheating Units

Fig. 8 shows the steam-te perature control characterist of typical resuperheating un of three types for the 1450-p Br cycle. Section (*a*) describ the behavior of a unit such shown in Fig. 4, includi a combination radiant-conve tion superheater and a co vection resuperheater. In sp of the naturally steep load-fu nace exit temperature chara teristics of furnaces, the fi primary-steam-temperatu curve of such a unit is ve flat over a very wide load ran because of the mutually co pensating characteristics of t radiant and convection sectio In the radiant section the stea temperature increases suf ciently to compensate for t steam-temperature decrease in the convection section as the lo decreases. The net result is that little or no control is require The convection resuperheater is designed for the minimum lo at which the full turbine-load resuperheater steam temperature required. As the load increases, the temperature tends to i crease above the desired value and must be kept down by b passing gas around the resuperheater or by desuperheating wi a water spray.

Section (*b*) shows the behavior of a unit such as shown in Fig. with superheating and resuperheating sections only of the co vection type designed for steam-temperature control down to 7 per cent of full load with standard firing, and section (*c*) for di ferential firing. For the same control range, the unit designed f differential firing requires less superheating and resuperheatin surface and less total pressure surface for the same efficienc However, there is a slight penalty for this steam-temperature co trol by furnace-temperature control, reflected in a loss in efficienc at partial loads, because the increase in furnace temperature pr duced to raise the steam temperature is reflected in an increase i exit temperature, as compared with that attained in a unit wit standard firing for the same control range. In twin-furnace unit with differential firing, the increase in temperature in one furnac is nearly compensated for by a decrease in the other, and the effe on unit exit temperature and efficiency is practically nil.

Since any change that must be effected in the temperature o the primary steam reacts on the temperature of the resuperheate

team also, the problem of control is the more difficult the more each temperature has to be controlled separately. In this respect, units with radiant superheating sections are the least troublesome because, as may be seen from section (a) Fig. 7, practically no control is required to maintain the primary-steam temperature constant over a wide load range. By using resuperheaters exclusively of the radiant type, the control problem is further simplified because the resuperheater duty imposed by the cycle and the resuperheating capacity of a radiant superheater both increase with decreasing load and are approximately complementary.

Resuperheating Units With Radiant Resuperheaters

While it might have been of interest to extend the economic comparisons of this paper to include radiant resuperheaters, since a number of such units, as indicated in Table 2, have been in satisfactory use, it seemed undesirable to enlarge the scope of this discussion at this time.

Operating Problems

While operating limitations of resuperheating units imposed by the slagging characteristics of the ash in the fuel are more severe, and starting and control problems somewhat more difficult, in other respects properly designed resuperheating units are no more troublesome to operate with equal reliability and availability than nonresuperheating units for the same primary steam conditions.

The design of resuperheaters for long life and trouble-free behavior requires considerable care because of the limited resuperheated steam-pressure drop allowable. This, for the cycles considered, is about 20 psi at full load. Because of this the arrangement, design, and operation of the burners require particular care in order that the gas flow, heat, and temperature distribution over the entire surface be as nearly uniform as possible.

Since the steam-flow rates in the elements are quite well determined for each design by the allowable steam-pressure drop, and the metal temperatures at these steam-flow rates by heat-absorption rates, gas, and steam temperatures, adequate alloys must be used in each section to suit the conditions. These metal selections and their proportions have a marked influence on the cost and life.

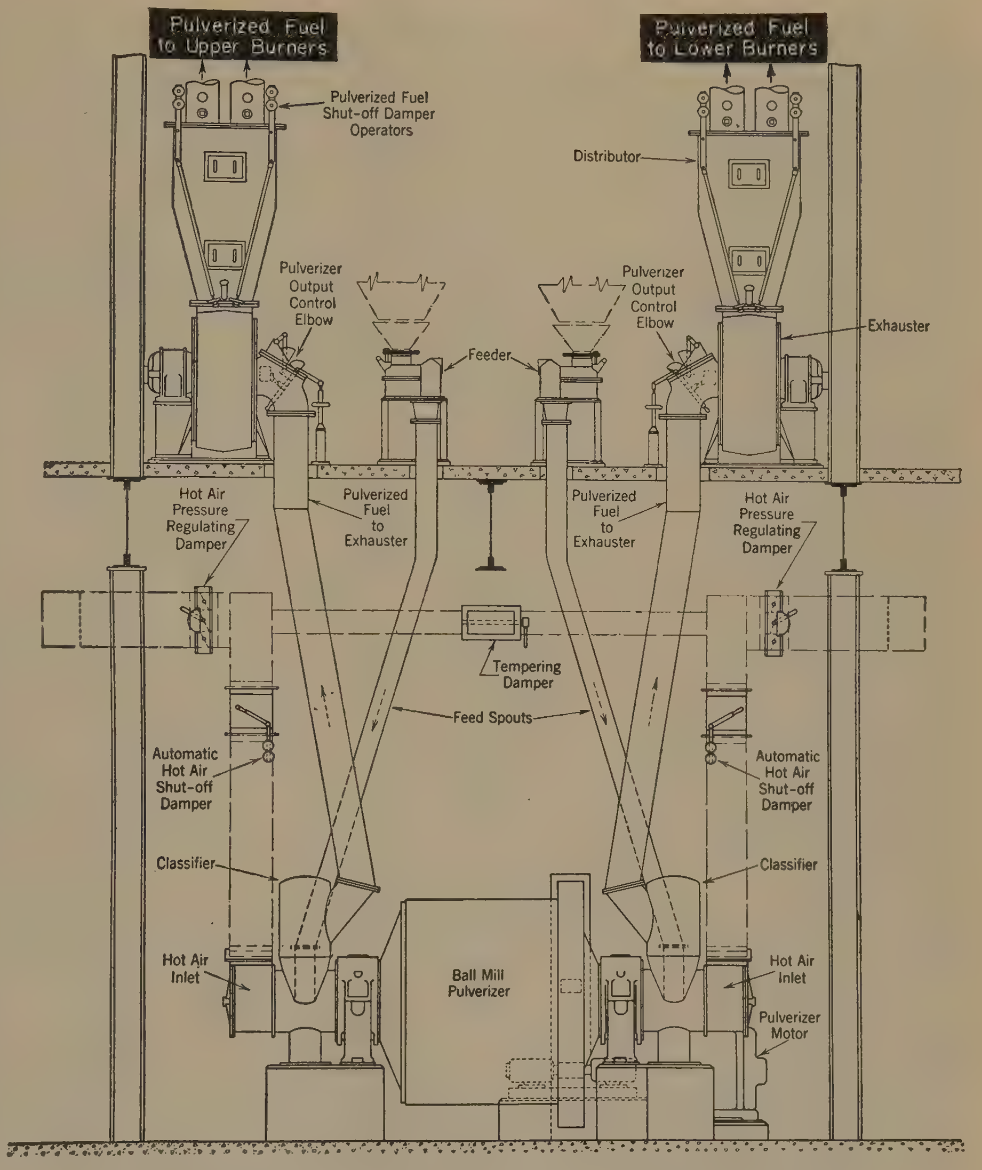


Fig. 6 Twin-Exhauster Ball-Mill Arrangement for Differential Firing

Conclusions

1 Steam generators for turbine generators employing resuperheating cycles cost little if any more and may, for a small steam-temperature control range, cost less than steam generators for nonresuperheating units for the same turbine capability and primary-steam conditions.

2 Resuperheating units with superheaters and resuperheaters entirely of the convection type are more difficult to design for low furnace-exit temperatures and are more restricted as to choice of fuel than the nonresuperheating units.

3 Resuperheating units with radiant superheaters and resuperheaters are not restricted as to choice of fuel, and while more expensive for small steam-temperature-control ranges and high furnace-exit temperatures, they become cheaper than units of the exclusively convection type for furnace-exit temperatures under about 1960 F for the 1450-psi, and about 2130 F for the 2000-psi cycle.

4 While operating limitations of resuperheating units, imposed by the slagging characteristics of the ash in the fuel are more severe, and starting and control problems require more care, resuperheating units require little more skill to operate than non-superheating units for the same primary-steam conditions.

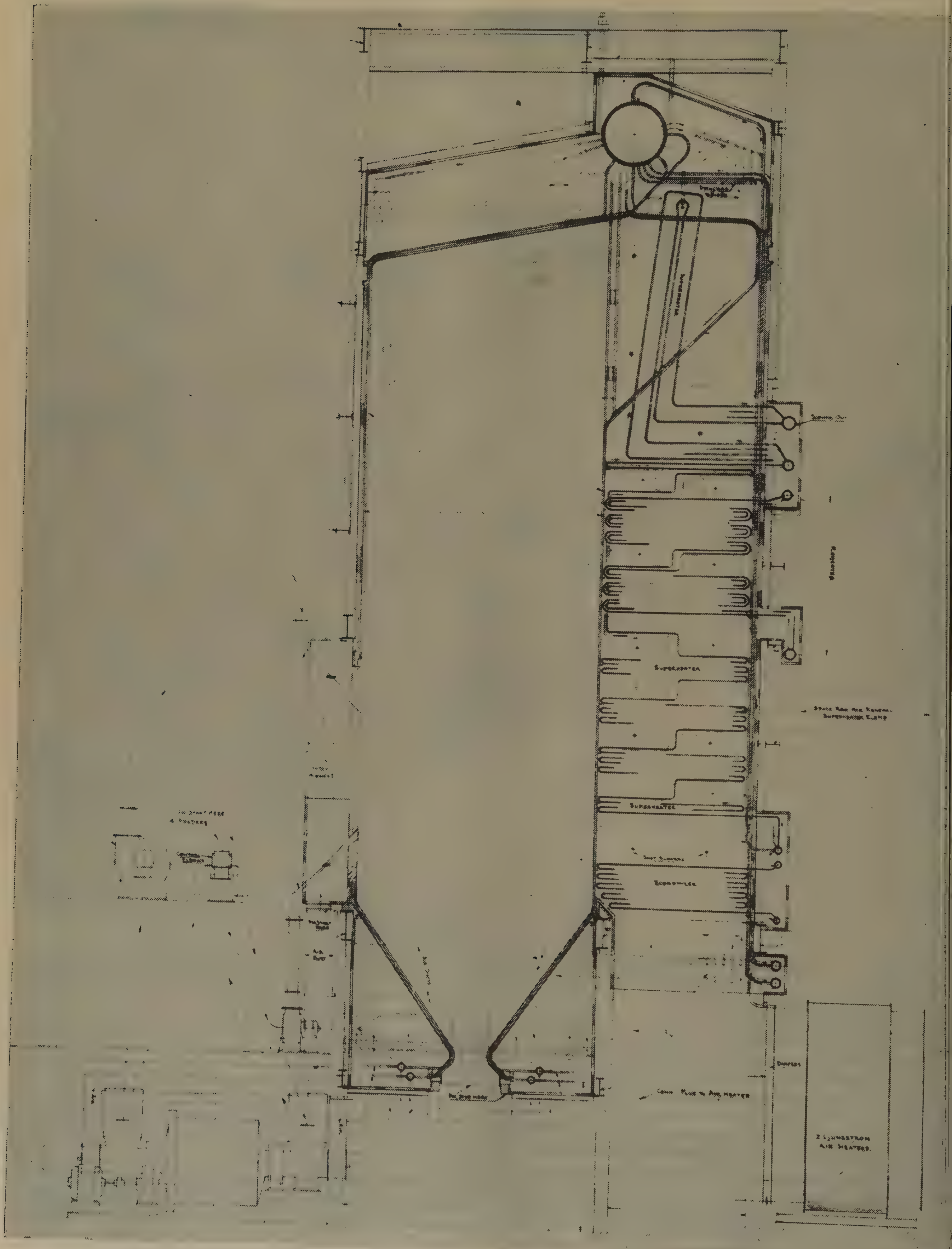


Fig. 7 Typical Resuperheating Steam Generator With Convection Superheater and Convection Resuperheater

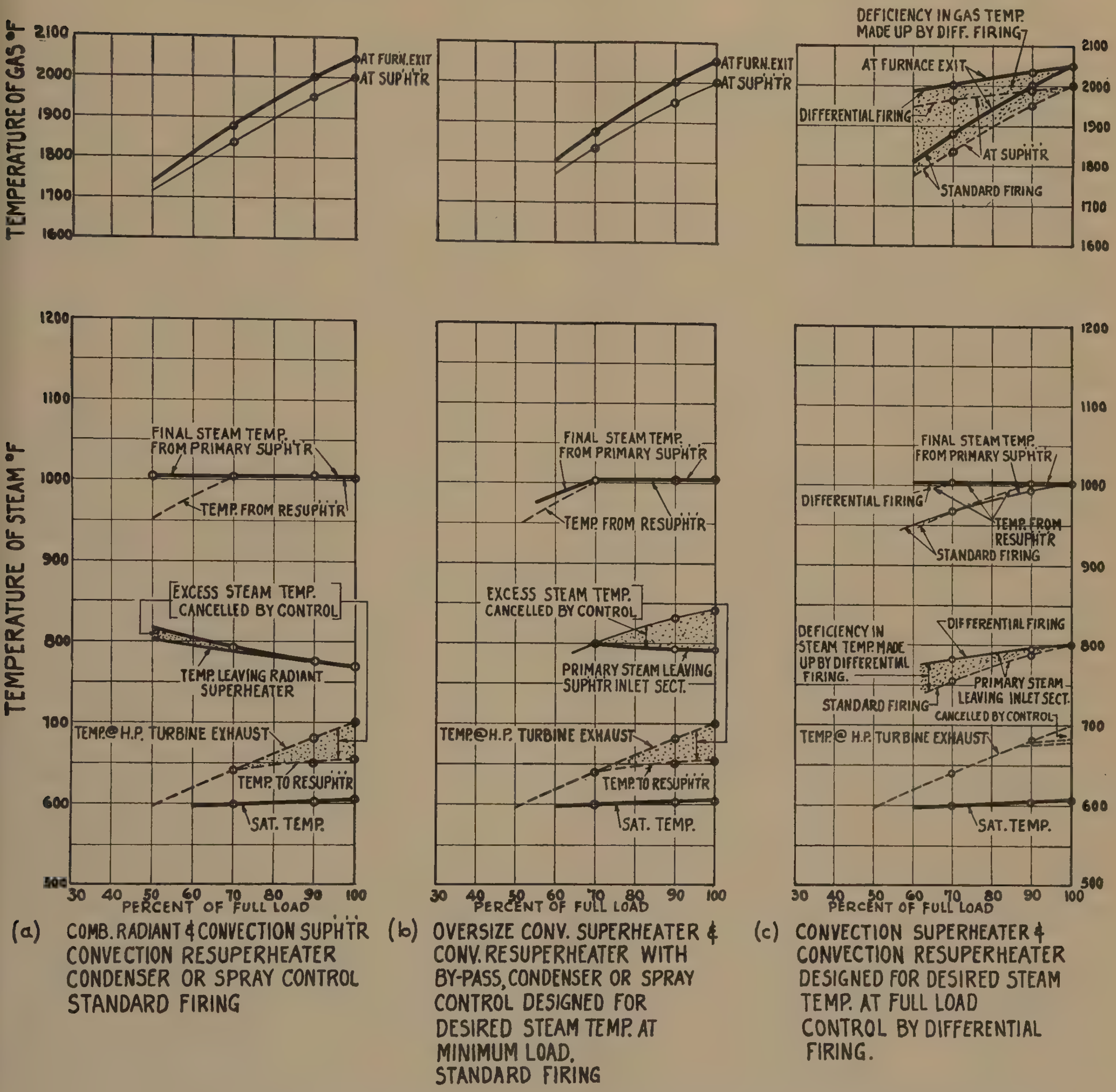


Fig. 8 Control Characteristics of Resuperheating Steam Generators With and Without Radiant Superheaters

5 Secondary but important effects, tending to reduce the first cost of resuperheating plants, is the reduction in the capacity of fuel-handling and storage facilities, feed pumps, heaters, evaporators, condensers, and condenser auxiliaries. The auxiliary power for coal-handling, preparation, and pulverizing, and in pumping water, air, and flue gas will be less, too, per kilowatt hour generated, and station sendout per kilowatt turbine capability will be greater.

High-Pressure Boilers With Reheaters

By W. H. Rowand,[1] A. E. Raynor,[2] and F. X. Gilg,[3] New York, N. Y.

When power generating equipment is designed to operate within the reheat steam cycle, the steam generating unit includes a reheat section in which partly expanded steam, returned from the turbine, is reheated before final expansion takes place. Reheat boiler units which have been in service for more than 20 years are described, together with some modern high-pressure high-temperature units. Comparison of reheat with nonreheat steam generating units to generate the same kilowatt capacity at the same boiler efficiency and draft loss is shown, together with estimated cost increase of the reheat boilers.

In THE design of new equipment to produce electrical energy, engineers are searching constantly for improved steam-cycle efficiency which will produce a kilowatt from fewer heat units to offset rising costs. The use of higher steam pressures and temperatures improves steam-cycle efficiency. The application of the reheat steam cycle also improves steam-cycle efficiency. The purpose of this paper is to describe the application of reheaters to modern high-pressure boilers.

In the reheat steam cycle, partly expanded steam is withdrawn from the turbine and led to a reheater in which the steam is reheated, i.e., its temperature and heat content are increased. From the reheater, the steam is led back to the turbine or another turbine to complete its expansion. As can be seen from a temperature-entropy chart, a greater portion of the heat added to the steam in the reheat cycle is converted to useful energy than would be possible in a normal steam cycle without reheat. Also, the moisture content of the steam in the low-pressure stages of the turbine is reduced so that destructive erosion of the turbine blades is minimized.

When a high-efficiency steam generating unit is available as part of the equipment in a reheat steam cycle, the net heat rate in Btu required to produce a kilowatt of electrical energy is reduced. A major factor in operating cost is reduced.

Boilers equipped with reheaters for heating steam, after it has given up some of its energy through partial expansion in a turbine, have been in service for more than 20 years. In the mid-twenties, pioneering in steam pressures progressed very rapidly from the 450-psi level to 1200 psi. Available materials and designs, however, limited steam temperatures to 750 F, and this temperature limitation, with the straight-expansion cycles then in common use, imprisoned much of the inherent advantage of higher steam pressures. Reheating offered a solution.

Obviously, the theoretical heat conservation from reheat had to be very carefully guarded against the rapid inroads of costs for design, construction, and operation of the reheat equipment, but the efforts of the station designers and of the manufacturers finally produced a balance clearly in favor of reheat.

With the development of alloy materials suitable for higher temperatures, it became more profitable to invest in the nonreheat system with higher primary-steam temperatures, and to give up the relatively less productive savings from reheat. Now we are again reaching a temporary limitation in steam temperatures, and with the rapidly increasing cost of fuels, the reheat cycle is again becoming popular, but at a much higher level of temperature and pressure than that at which it was attractive in the mid-twenties. Reheater units are being built today for primary-steam temperature of 1050 F, reheat-steam temperature of 1000 F, and initial pressures up to 2500 psi.

The use of higher steam pressures and temperatures has resulted in a steady reduction of station heat rates on new generating equipment. There does not seem to be much probability of using pressures higher than 2500 psi. The possibility of future improvement lies more in the direction of higher temperatures, but, for the time being, 1100 F is about the top commercial limit to high temperatures at high pressures. Therefore utility engineers, in their relentless pursuit of lower costs in generating power are again considering the reheat cycle at temperatures of 1000 F and even 1050 F, and are finding that, in some cases, it is economically justified in the light of present-day fuel costs.

Reheaters are actually superheaters and are generally installed as convection surface in a boiler setting. In the mid-twenties, a few live-steam reheaters were used. Reheaters are subject to the same design factors as superheaters, requiring the correct amount of surface, arranged with the proper internal and external flow areas to insure satisfactory tube-metal temperatures and a uniform distribution of steam and gas flow over the reheating surfaces. Actually, reheater design is more difficult because of the necessity of designing for a minimum pressure drop in the reheater and the connecting piping. The surface should be installed in a zone where the temperature of the gas is high enough to insure economical use of the heating surface and low enough to insure freedom from tube slagging.

Reheater Installations

In the early installations, reheaters were installed above or behind the boilers. The unit shown in Fig. 1 was designed and built about 1925. The reheater is located in the upper part of the boiler setting, between the boiler tubes and the horizontal circulating tubes. The boiler and superheater deliver steam at 1050 psi and 710 F to the high-pressure turbine from which partly expanded steam of 365 psi and 505 F is returned to the boiler for reheating to 700 F, from where it is discharged into the 350-psi header serving the low-pressure turbine. A guillotine damper, movable in a vertical direction, regulates the steam temperature leaving the reheater by varying the amount of gas passing over the reheater surface. Similar but larger steam generators with reheaters were installed in this plant in 1929.

The unit shown in Fig. 2 was built about 1928, and is typical of several other units built about that time. The reheater was installed above and to the rear of the boiler and was of sufficient size to reheat the steam supplied to the turbine by three boilers generating steam at 730 psi and 750 F. Partly expanded steam entered the lower drum of the reheater section at 190 psi and 490 F. The temperature of the steam leaving the reheater was raised to 760 F.

In each of these examples of the early application of reheaters, the design of the boilers was such that the reheater surface was traversed by flue gas having a temperature of about 1200 F.

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[3] Application Engineer, The Babcock & Wilcox Company. Mem. ASME.

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Obviously, less surface would be necessary and higher steam temperatures possible, if the reheater surface could be installed in areas where the gas temperature is higher. As will be shown by subsequent illustrations, the design of modern high-pressure high-temperature high-capacity boilers to operate with high availability, lends itself much more readily to the application of reheater surface in zones where the gas temperature is much

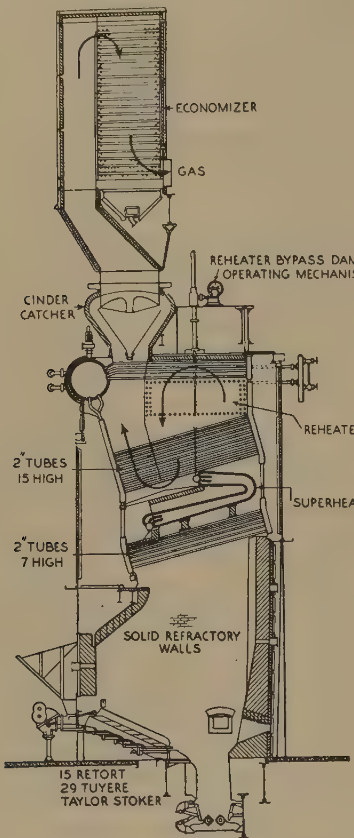


Fig. 1 High-Pressure Boiler With Reheater—1925

higher, resulting in a more economical use of the installed surface.

Another favorable factor, which developed in the intervening years, is the more general use of the single-boiler single-turbine arrangement, which results in much simpler piping connections between the turbine, superheater, and reheater.

A more recent application of reheaters is shown in Fig. 3, a pulverized-coal-fired slag-tap unit in the Twin Branch Station.[4] This unit is designed to generate 550,000 lb of steam per hr at 2300 psi and 940 F primary steam conditions, with the partly expanded steam reheated at 400 psi to 900 F. The reheater surface of this unit is in three sections. The first section is a downflow convection section from header H to header J, located just above the screen tubes protecting the entrance to the convection section. The reheater section in this area occupies about $^1/_4$ of the width from side wall to side wall, the remaining $^3/_4$ being occupied by a similar tube bank of the primary superheater. The second section of the reheater surface forms the side walls of the first and second open passes, from header K to header L. The steam flows upward and collects at the top in a loop header M from which the reheater tubes emerge to form the

[4] "Operating History of the 2500-Psi Twin Branch Plant," by Philip Sporn and E. G. Bailey, Trans. ASME (Supplement Section—Furnace Performance Factors), vol. 66, 1944, 16 pp.

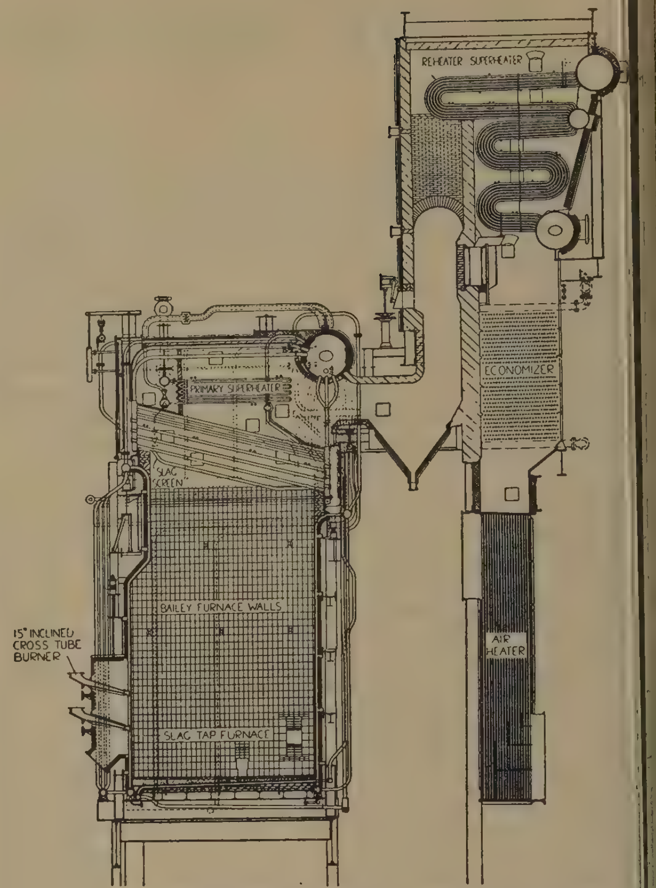


Fig. 2 High-Pressure Boiler With Reheater—1928

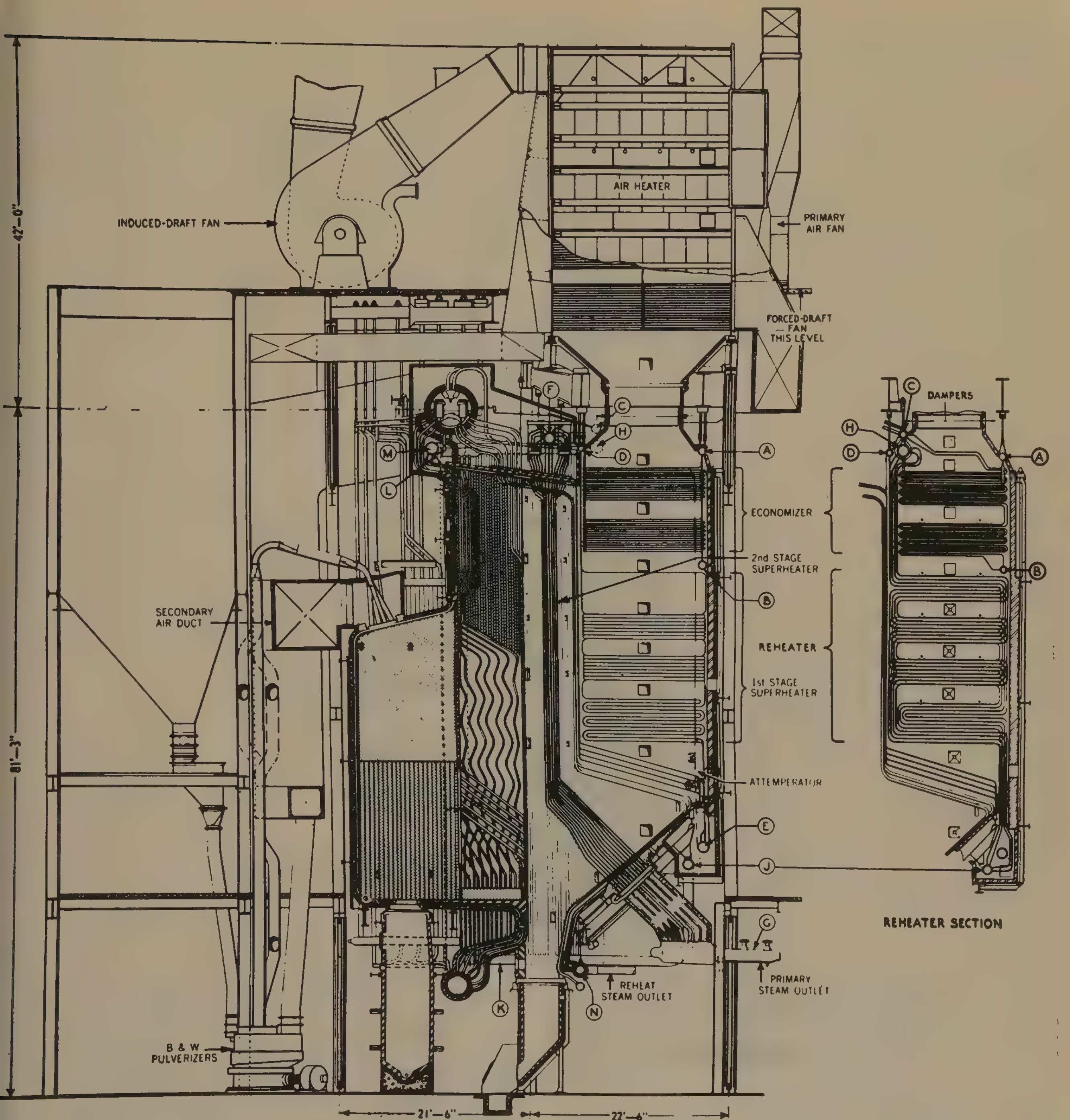
roof over the open pass and thence downward, behind the open-pass baffle-wall tubes to form the screen, and thence under the hopper-floor tubes to an outlet header N.

The first section is located in a convection zone, and the other two sections are located in areas where radiant heat absorption is predominant.

This unit has been operating successfully for more than 7 years. It operates at maximum load with yearly outages. Its availability has been in the order of 95 per cent. The station heat rate of the unit served by this reheat steam-generating unit is approximately 10,000 Btu per net kw.

At these higher temperatures the control of the steam temperature is important from the standpoint of maintaining turbine efficiency and avoiding excessive metal temperatures in the turbine, superheater, or reheater. In this unit, designed for 550,000 lb of steam per hr, the primary steam temperature is controlled by means of a spray attemperator, spraying high-pressure feedwater in the steam leads between the primary and secondary sections of the superheater. The steam temperature from the reheater is controlled by varying the gas flow over the convection section of the reheater surface by means of dampers at the outlet over the reheat section of the unit.

Experience gained on this unit has permitted a simplification of the arrangement of surfaces in later designs. Fig. 4 shows a larger unit for the same station, designed to generate 930,000 lb of steam per hr at 2080 psi and 1050 F, and to reheat 835,000 lb of steam per hr at 425 psi from 662 F to 1000 F. The steam-generating unit is designed to operate with an efficiency of 89.3 per cent. The combination of this higher boiler efficiency with the use of the reheat steam cycle with regenerative feedwater



ECONOMIZER FLOW
A INLET HEADER
B INTERMEDIATE HEADER TO BY-PASS SECTION
C OUTLET HEADER FROM BY-PASS SECTION

SUPERHEATER FLOW
D FIRST STAGE INLET HEADER
E " " OUTLET "
F SECOND " INLET "
G " " OUTLET "

REHEATER FLOW
H FIRST STAGE INLET HEADER
J " " OUTLET "
K SECOND " INLET "
L " " OUTLET "
M THIRD " INLET "
N " " OUTLET "

FIG. 3 HIGH-PRESSURE BOILER WITH REHEATER—1941
(550,000 b per hr, 2300 psi, 940 F, 900 F reheat. Twin Branch Station.)

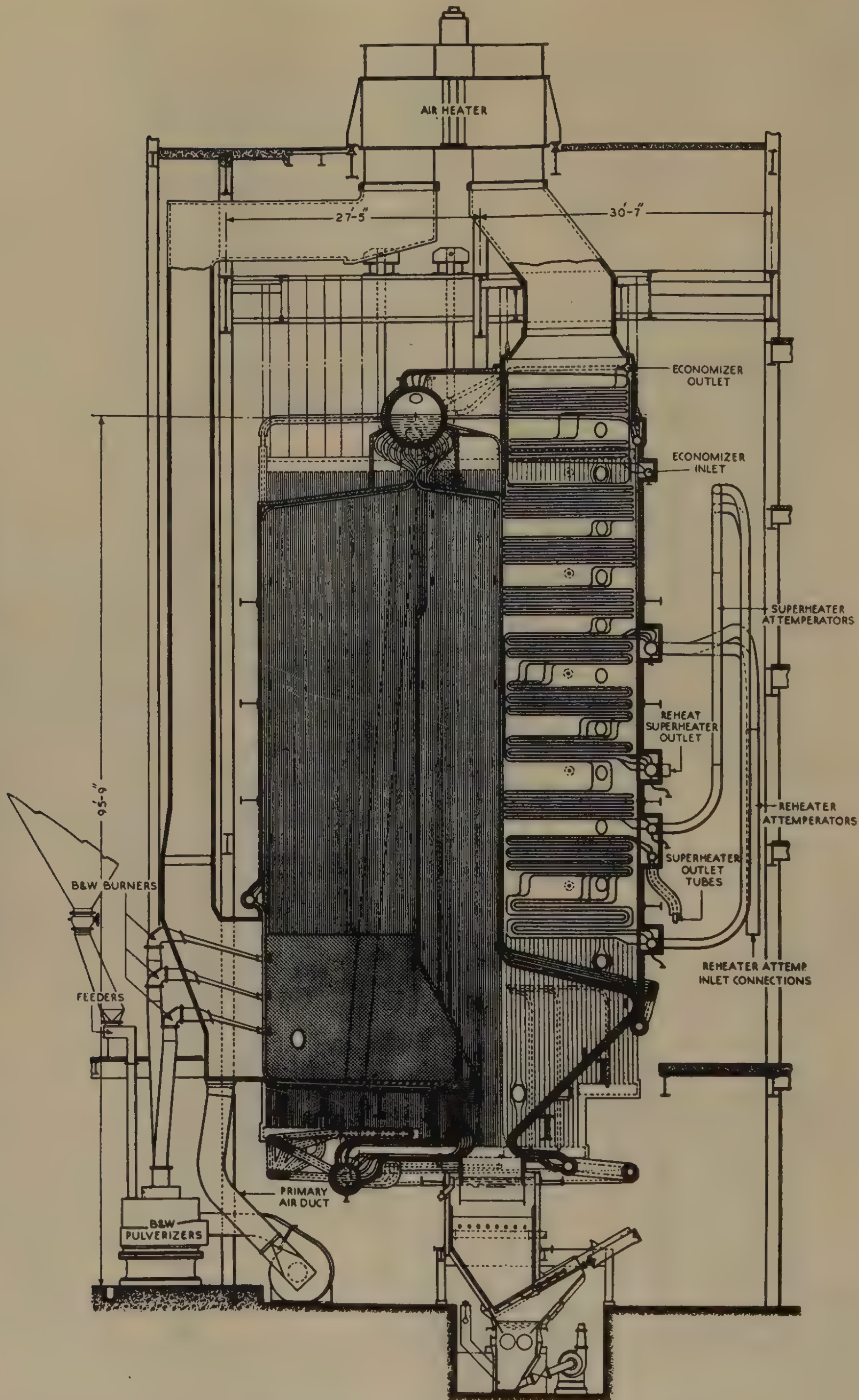


Fig. 4 High-Pressure Boiler With Reheater—1948
(930,000 lb per hr, 2080 psi, 1050 F, 1000 F reheat. Twin Branch Station.)

heating will result in a station heat rate of about 9300 Btu per net kw.

All the superheater and reheater surfaces are located in the convection section of the unit, the reheater being installed between two sections of the high-pressure superheater.

With primary steam temperatures of 1050 F and reheat steam temperature of 1000 F, it is necessary to have the most accurate control available for steam temperature and reheat temperature for reliability. Spray attemperators are being used because of the successful experience over a period of 7 years with the un shown in Fig. 3, and many other high-temperature units. Th spray atttemperator for controlling primary steam temperatu is located between two sections of the high-pressure superheate The attemperator for controlling reheat steam temperature i located in the inlet pipe of the reheater.

Fig. 5 shows diagrammatically, the design of spray attempera tor used in these installations. Feedwater is introduced into th steam pipe through an efficient atomizing spray nozzle. Th

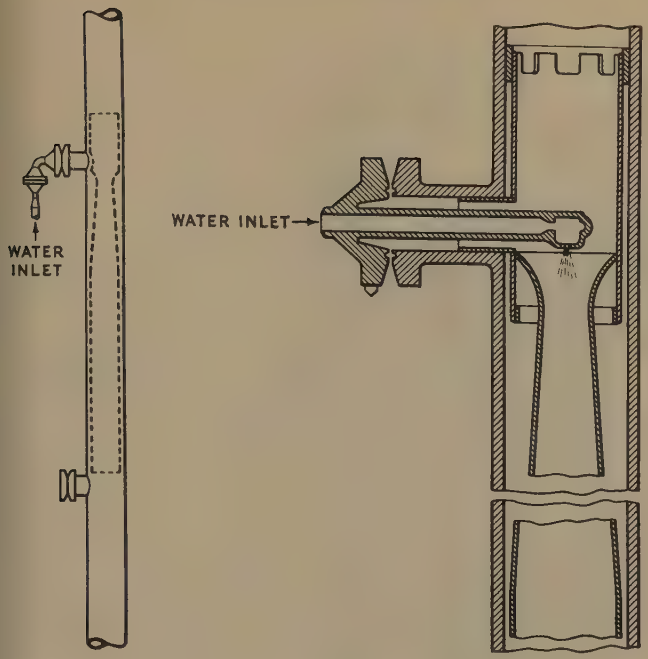


FIG. 5 SPRAY ATTEMPERATOR

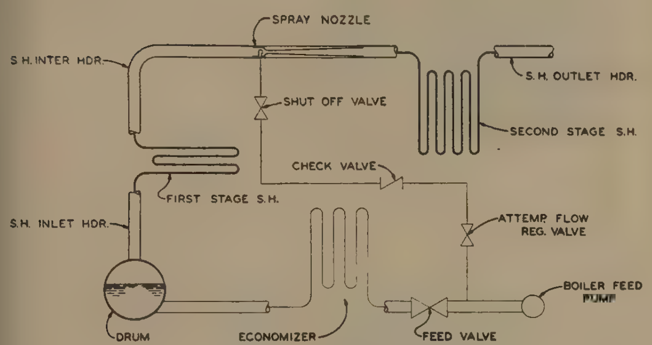


FIG. 6 SCHEMATIC CONTROL DIAGRAM OF SINGLE-NOZZLE SPRAY ATTEMPERATOR

atomized spray is mixed effectively with the steam by a Venturi-shaped liner, insuring rapid evaporation and superheating of the spray water. The liner prevents any spray water from reaching the pressure piping, thus eliminating any possibility of temperature shock and the quench cracking that has occurred in some unlined steam pipes.

Fig. 6 shows the diagrammatic arrangement of the spray-attemperator control and piping. The pressure drop across the feedwater-regulator valve, economizer, and primary section of the superheater is utilized to force the feedwater through the attemperator pipe and spray nozzle. Temperature elements in the steam lines automatically control the amount of water passing through the regulator valves. To further assure instantaneous response, the control relays are locked with steam-flow or air-flow impulse. This arrangement assures close control of temperature at the desired level throughout the design load range for constant temperature.

Fig. 7 shows the design of units being built for the Philip Sporn Station,[5] shown in Fig. 8. The capacity and steam conditions are the same as for the unit in Fig. 4 but the furnace is designed for dry-ash removal. The convection section will be very similar to the convection section of the slag-tap unit in Fig. 4.

In order to increase the steam-temperature control range, these units are being designed to recirculate flue gas from the economizer outlet at partial load back into the furnace adjacent to the burners. This will have the effect of decreasing the radiating temperature and absorption in the furnace, thereby increasing the heat content of the gases to the superheater and reheater.

Fig. 9 shows a modern radiant-type boiler designed to generate 575,000 lb of steam per hr at 1500 psi and 1000 F, reheating 502,000 lb of steam per hr at 417 psi from 697 F to 1000 F. The outlet superheater tubes and the location of the outlet header are arranged with sufficient flexibility to accommodate the entire expansion of the piping. The reheater is located in the intermediate gas-temperature zone between two sections of the superheater. The reheater outlet is also arranged to accommodate the expansion of the piping to the low-pressure turbine. Crossflow of the gases is utilized throughout the superheater and reheater to obtain optimum absorption efficiency of the heating surface. On this type of unit, considerable temperature adjustment can be obtained by using different combinations of burners at different loads.

Fig. 10 shows a pulverized-coal-fired slag-tap unit for Oswego Station, designed for 655,000 lb of steam per hr at 1450 psi and 1000 F, reheating 575,000 lb of steam per hr at 405 psi from 701 F to 1000 F. Primary and reheat steam temperatures will be controlled to 70 per cent of full load with spray attemperators between the primary and secondary superheater sections and at the inlet to the reheater. The same furnace-design factors were used on this job as were used on the similar nonreheat unit in the same station which has been in operation several years.

Fig. 11 shows a cyclone-furnace-fired slag-tap unit for Waukegan, designed for 830,000 lb of steam per hr at 1850 psi and 1010 F, reheating 733,000 lb of steam per hr at 550 psi from 690 F to 1010 F. Primary and reheat steam temperatures will be controlled by spray attemperation down to 65 per cent of full load. The temperature-control range is further extended to less than half load by recirculating gases from the economizer outlet to the furnace to reduce the radiating temperature, which in turn reduces the heat absorbed in the furnace and increases the heat content of gases to the superheater.

OPERATION

It is essential to design the boiler, turbine, and their interconnections so that they will have sufficient flexibility and safety during normal starting up and shutting down and also during emergency trip-outs and restoration to full-load conditions. The inclusion of reheaters introduces additional considerations to the co-ordinated design.

Fig. 12 shows diagrammatically the connections between the boiler and turbine for the units shown in Figs. 4 and 7. There is the usual automatic emergency stop valve in the high-pressure line between the boiler and turbine and the usual control valves on the high-pressure turbine. Two spring-loaded safety valves and power control valve are located at the superheater outlet. As required by the Code, safety valves, having a relieving capacity of 100 per cent of the capacity of the unit at 450 psi, are located on the lines between the high-pressure turbine and the reheater, and on the pipe between the reheater and low-pressure turbine. An automatic interceptor valve is located just ahead of the low-pressure turbine to prevent it from overspeeding in case the high-pressure stop valve trips. The customary drains are located on the superheater and reheater headers and steam piping.

The normal starting-up schedule, as worked out for these

[5] "The 2000-Psi, 1050 F, and 1000 F Reheat Cycle at the Philip Sporn and Twin Branch Steam-Electric Stations," by Philip Sporn, Trans. ASME, vol. 70, 1948, pp. 287–294.

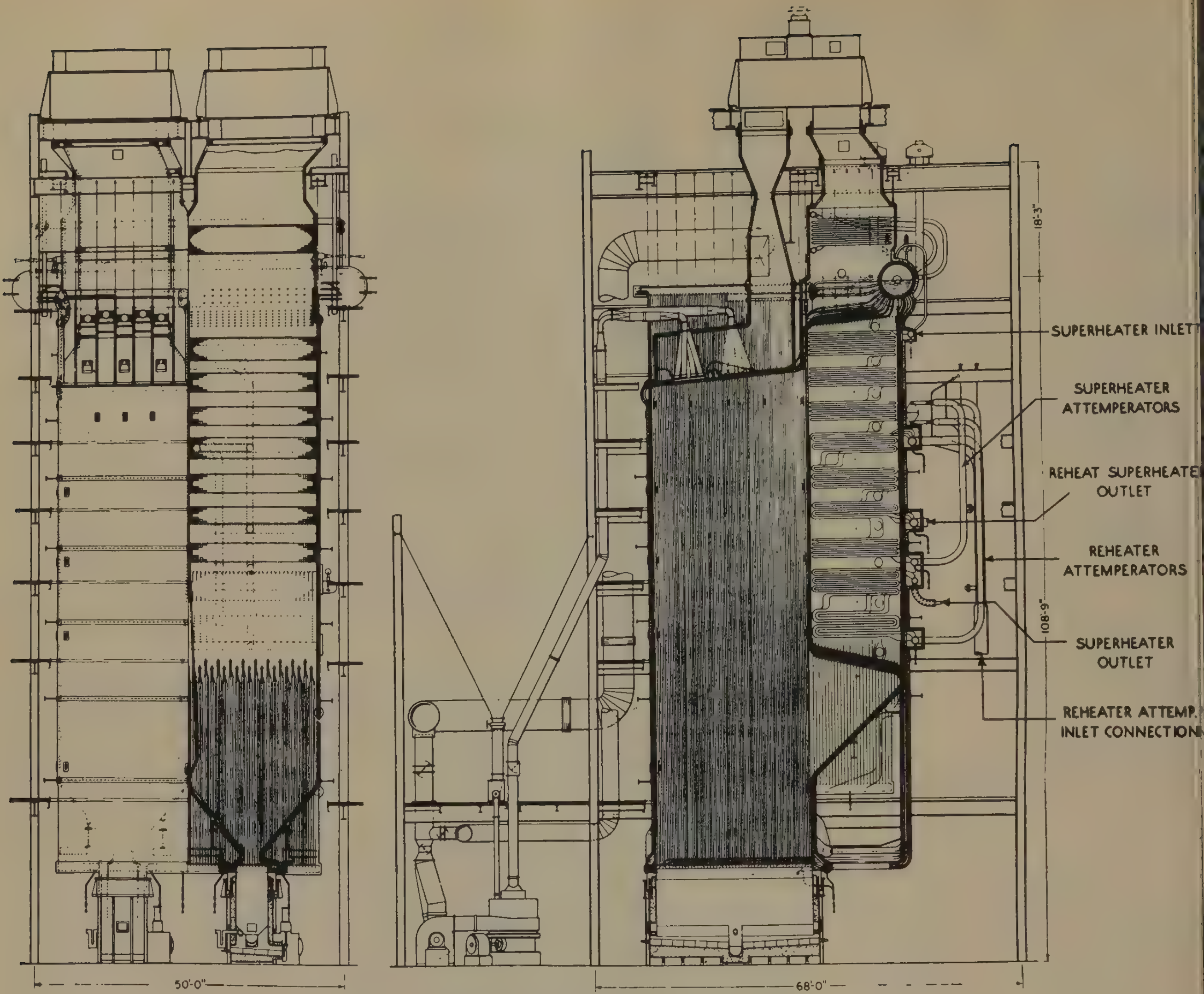


Fig. 7 High-Pressure Boiler With Reheater—1948
(930,000 lb per hr 2080 psi, 1050 F, 1000 F reheat. Philip Sporn Plant.)

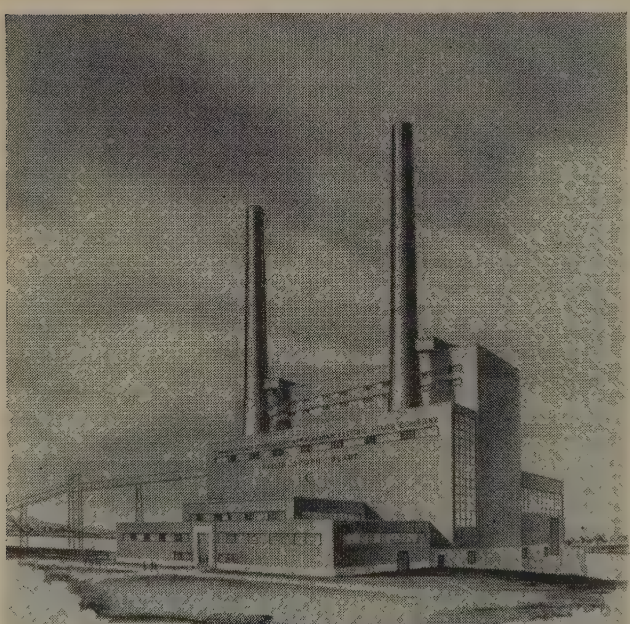


Fig. 8 Philip Sporn Plant

high-pressure units, calls for firing the boiler with pulverized coa and oil torches at low input, bringing the unit from cold to fu load and temperature in about 8 hr.

Another important consideration is the handling of the uni during an emergency trip-out and providing for adequate protec tion of the equipment and facilities for restoring the load a rapidly as possible. For the units shown in Figs. 4 and 7, wher the reheater is located in an intermediate gas-temperature zon between two sections of the superheater, a simple straightforwar method of handling the unit under these conditions has bee worked out, without having to restort to the complications o steam by-pass lines, reducing valves, and the like. The high pressure trip valve, the throttle valve, and the interceptor valv are each being provided with an additional contactor, which wi be wired through the interlock system to trip out one or mor pulverizers automatically after a time delay of a few seconds afte closing.

Boiler Comparison—With and Without Reheaters

The number of Btu required to produce a kilowatt at variou steam pressures and temperatures, with and without reheat, ind cates that the reheat cycle requires fewer Btu. The question the becomes: "What is the relative cost of the high-pressure high temperature unit without reheat as compared with the rehea unit for the same conditions?" This question has been considere many times. Boiler designs have been prepared for units with an

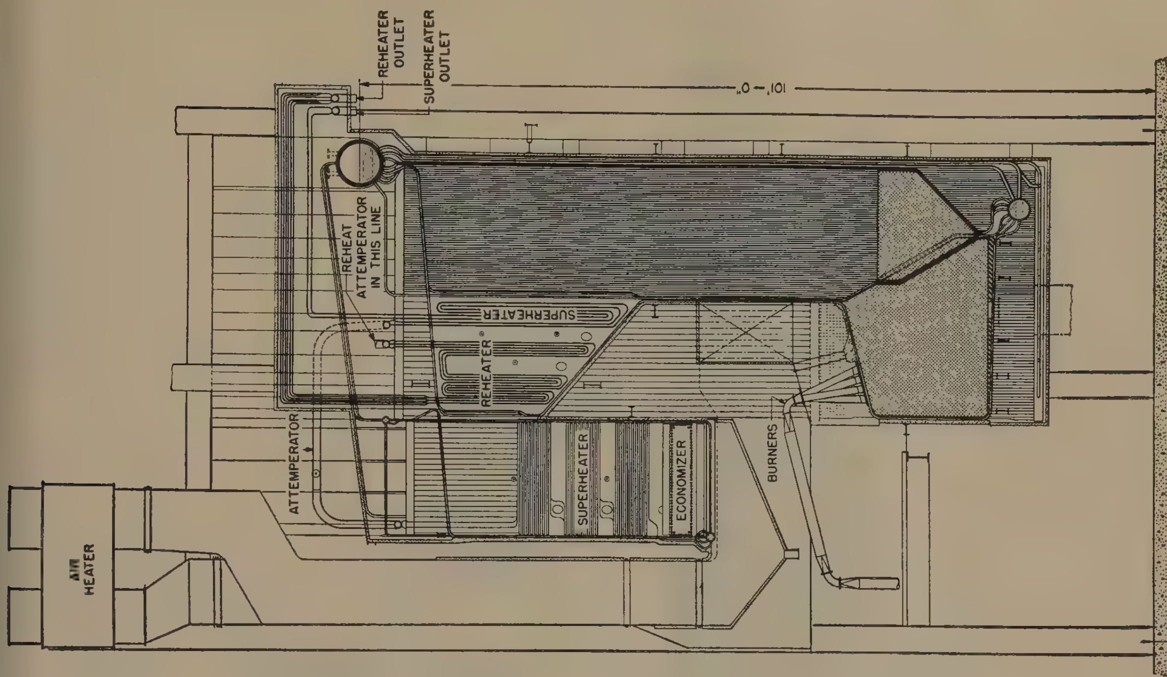


Fig. 10 Reheat Boiler—Oswego Station

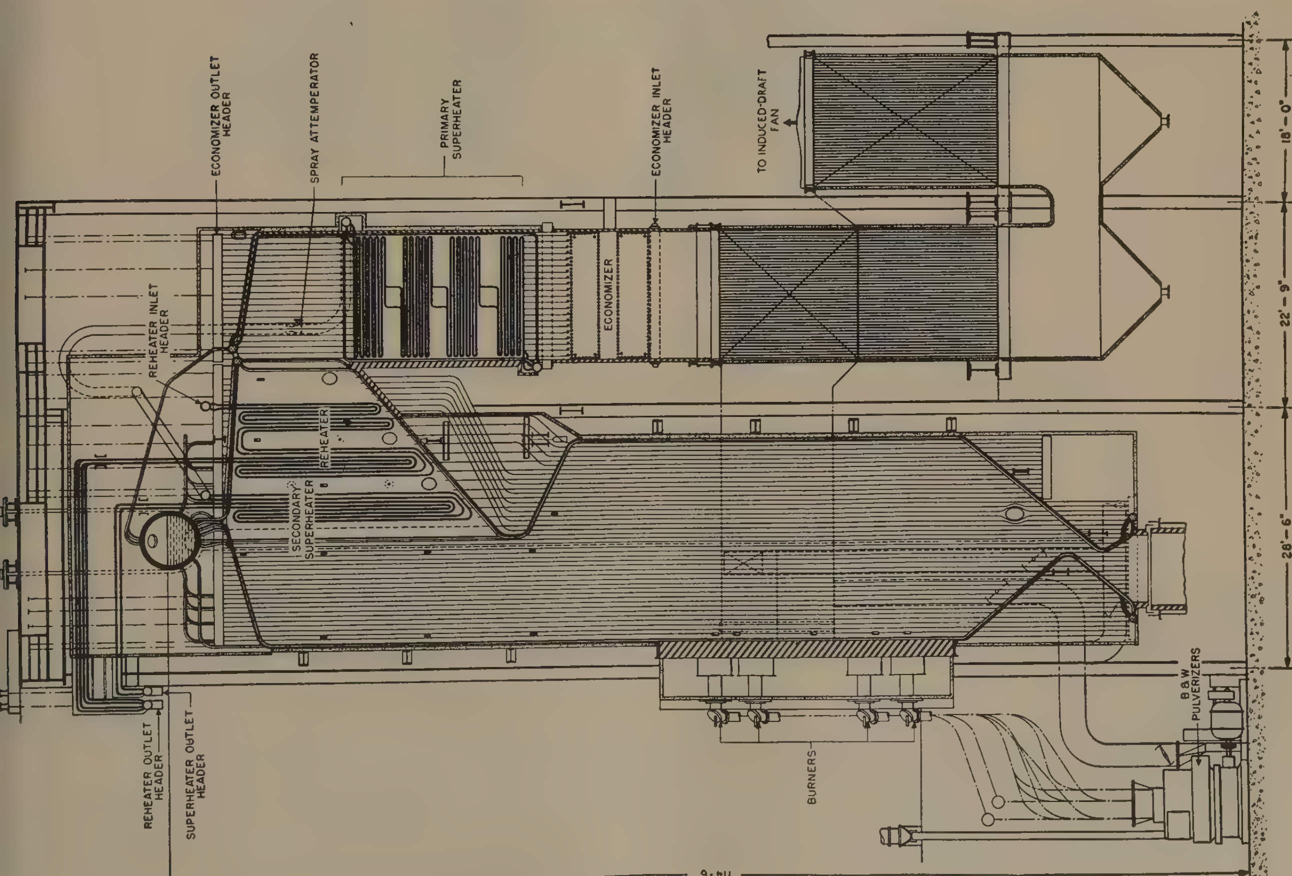


Fig. 9 (*left*) High-Pressure Boiler With Reheater—1948
(575,000 lb per hr, 1500 psi, 1000 F, 1000 F reheat. Salem Harbor Station.)

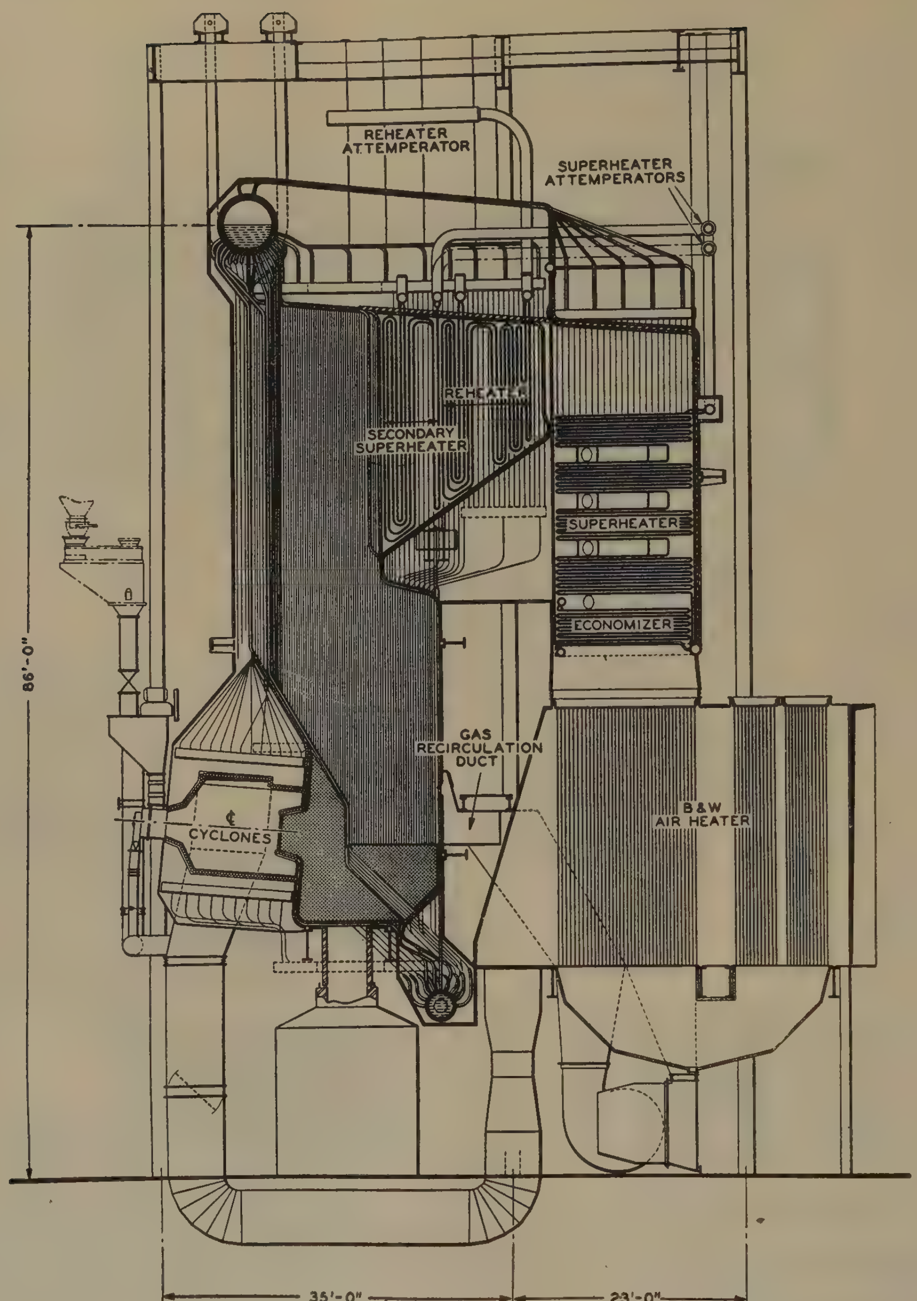


FIG. 11 REHEAT BOILER, CYCLONE-FURNACE-FIRED—WAUKEGAN STATION

without reheat to help arrive at the over-all cost of the two types of units. Several items of course must be considered in arriving at the economic answer, the cost of boiler, piping, turbines, and control equipment.

To get comparable answers on boiler costs, the designer must take into account the fact that the heat input and primary-steam output of the reheat unit will be less than those for the nonreheat unit when both are designed to produce the steam required to generate the same number of kilowatts from the same primary-steam conditions and at the same boiler efficiency and draft loss.

Therefore the width of the reheat unit will be smaller. The nonreheat unit will have more economizer surface to make up for the heat absorption of the reheat surface. Economizer surface, in general, is less costly than reheater surface because some al materials are required in the reheater.

The percentage distribution of economizer, superheater, a reheater heat absorption is indicated in Table 1, calculated fo reheat and nonreheat unit, designed for 1500 psi and 1000 primary and reheat steam temperature with 425 F feedwater te perature:

TABLE 1 HEAT-ABSORPTION DISTRIBUTION IN NONREHE AND REHEAT UNITS

	Nonreheat	Rehe
Heat absorbed in furnace and boiler, per cent	56.1	52.
Heat absorbed in superheater, per cent	29.4	25.
Heat absorbed in reheater, per cent	...	13.
Heat absorbed in economizer, per cent	14.5	8.
Total, per cent	100.0	100.
Heat absorbed in air heater, per cent	9.3	9.

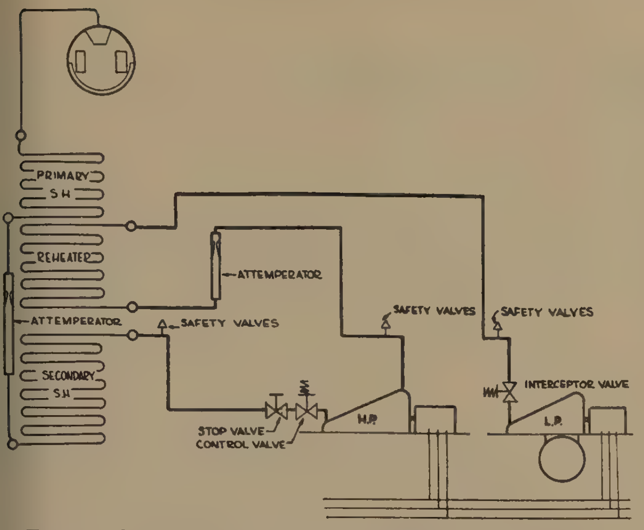


Fig. 12 Line Diagram—Boiler Turbine Arrangement

Design and price studies have been prepared for 60,000-kw, 80,-000-kw, and 125,000-kw units, some designed for liquid-ash removal, others for dry-ash removal. All of the units have overload capacities above these ratings. The steam requirements specified have not always been identical for the various sizes, but they have been reasonably close. Typical maximum-load steam requirements, specified by various companies for these units, have been about as indicated in Table 2.

A typical unit designed for dry-ash removal to satisfy the steam requirements of the 60,000-kw turbine is shown in Fig. 9, the 125,000-kw unit with a dry-ash-removal-type furnace is shown in Fig. 7, and a liquid-ash-removal-type unit is shown in Fig. 4.

The design and price studies made to date indicate that when the two types of units are designed to generate the same kilowatts, the reheat-boiler unit designed for 1500 psi and 1000 F costs about 5 per cent more than the nonreheat boiler. The difference in price between the reheat and nonreheat boiler is approximately the same for the 60,000-kw and the 80,000-kw units whether designed for liquid-ash removal or dry-ash removal.

TABLE 2 TYPICAL MAXIMUM LOAD STEAM REQUIREMENTS

Nominal turbine rating, kw	60000	80000	125000
Steam flow, lb per hr	575000	670000	930000
Pressure of steam at superheater outlet, psi	1500	1500	2080
Temperature of steam at superheater outlet, deg F	1000	1000	1050
Pounds of steam to reheater, lb per hr	502000	606000	840000
Pressure of steam at reheater inlet, psi	417	410	420
Pressure of steam at reheater outlet, psi	392	385	390
Temperature of steam at reheater outlet, deg F	1000	1000	1000
Temperature of feedwater to economizer, deg F	492	435	441

The reheat boiler designed for 2035 psi at the superheater outlet, 1050 F primary-steam temperature, and 1000 F reheat-steam temperature costs about 7.5 per cent more than the nonreheat boiler designed for the same superheater-outlet pressure and temperature.

In both instances the standard and reheat boilers are designed to produce the same number of kilowatts, with the same range of steam-temperature control, same boiler efficiency, and draft loss.

Conclusions

Reheaters have been in service for more than 20 years. Design and operating experience from these installations is available for application to modern steam-generating units for high pressure and temperatures. Five boilers of the Fig. 7 design, two boilers of the Fig. 9 design, and one boiler each of the Fig. 4, Fig. 10, and Fig. 11 designs are now being built. Many others are contemplated and in the design stages. Whether the added cost of the reheat cycle is economically justified depends upon a careful analysis of all the pertinent factors in each individual case. It is our opinion that the reheat cycle is practical from the standpoint of design and operating reliability.

Modern Reheat Boilers

By W. S. PATTERSON,[1] NEW YORK, N. Y.

This paper gives a brief historical résumé of reheat rogress followed by a discussion of the design factors hich dictate the size, shape, and proportions of a large re- eat steam generator. A few typical installations are llustrated and discussed, and general comments on opera- ional procedures are included.

Progress in Reheat Boilers

IN the earliest applications of the reheat cycle in power stations, dating back more than 20 years, reheating of the steam was done in separate reheaters. Steam was generated t 550 psi and superheated to 750 F in a conventional boiler unit, hen expanded through a turbine to 105 psi, reheated at that ressure to 700 F in a separately fired reheater, and then ex- anded in a condensing turbine. Some of the early reheat installa- ions used steam-to-steam reheaters and others used a two-stage rrangement combining gas-to-steam and steam-to-steam.

A few years later units were built with the reheater located vithin the confines of the steam generator, and some of these nstallations employed radiant reheat surface. Several such nits are in satisfactory operation, and in a few cases dupli- ate units were installed at a later date. Obviously, extreme are must be exercised to protect the reheater during the start- p, but in spite of this, some such units have been cross-con- ected, with two reheat units supplying a single turbine, and vith the necessary control equipment to proportion the return team to the two reheaters in a satisfactory manner.

Some reheat installations have been used in connection with 'topping" turbines in which the primary steam is expanded hrough a new high-pressure turbine and then reheated at the riginal pressure of the station for use in the older turbines. Such plants usually have a tie-in line permitting steam from low- ressure boilers to be used for reheater protection during starting p. They also have a by-pass around the high-pressure turbine, vith desuperheaters, to permit operation of the high-pressure oilers when the topper is out of service. This arrangement per- nits an emergency outage of the "topper" to occur without re- uiring the high-pressure boiler to be taken out of service, the team from the boiler first being passed through a reducing alve, then desuperheated, then reheated and delivered to the ow-pressure turbines. However, cross-connection of such units is xpensive and introduces operating complications (1–13).[2]

The high cost of fuel, material, and labor since the recent war uite naturally has increased the popularity of reheat boilers in entral stations because of better station economy, and many uch units are now in process of design or construction, selected for rimary-steam pressure of 1400–2200 psi, primary-steam tempera- ure up to 1050 F, and reheated-steam temperature up to 1000 F. The high cost of material and labor also makes it desirable to build hese units in large sizes, connected to a single turbine, and without ross-connection between the boilers. That this is being done is a tribute to the designers of boilers and turbines, but past per- formance is available to demonstrate that high availability can be built into both these pieces of equipment when proper attention is given to details of design and materials. Single-boiler reheat in- stallations are being considered up to a capacity of 1.5 billion Btu per hr fuel-firing rate.

Design Factors

When the specified primary-steam temperature is 1050 F and reheat temperature is 1000 F the combined superheater heat ab- sorption is about 40 per cent of the total heat absorption by all pressure parts, including furnace, boiler, and economizer. It may be said truly of such units that the rest of the installation is really built around the superheater and reheater.

The design is further influenced by the specified load range over which the steam temperature must be held constant. It is characteristic of a convection-type superheater that the outlet steam temperature increases as the load increases, and this in- crease may amount to 75 deg F from half to full load. Therefore, if full steam-temperature conditions are specified at reduced load, a control must be furnished to keep the steam temperature from rising. The primary cause of increased steam temperature at in- creased output is increased gas temperature.

When high primary and reheat steam temperatures are to be obtained by convection heat transfer, and with coal firing, the gas temperature leaving the furnace will be close to the fusion tem- perature of the ash, even at reduced load. Obviously, an increase in output with fixed burners will therefore produce two objec- tionable effects, namely, excessive furnace gas temperature and excessive steam temperature. Therefore it is doubly important to control the cause rather than the result of these increased tem- peratures.

With tilting burners, which in effect give an adjustable furnace, this is quite readily accomplished. The furnace can be designed to give the required furnace gas temperature at maximum load with the burners horizontal, and they are tilted upward at lower loads to control the gas temperature to exactly that required to give constant steam temperature.

To illustrate this design problem, Fig. 1 is a typical study made for a large unit similar to that illustrated in Fig. 5 in which all superheating and reheating surface is located beyond the furnace outlet, with the reheater located between two sections of the primary superheater. Curve *A-B* is the gas temperature entering

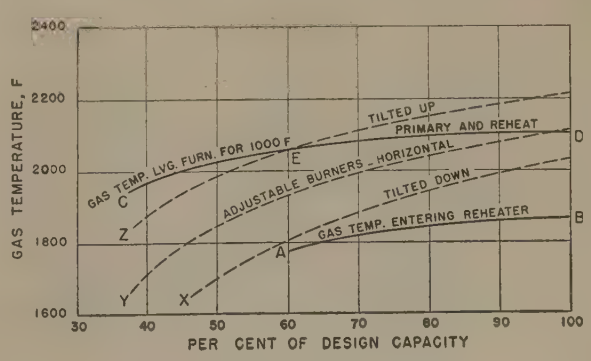


Fig. 1 Application Analysis Showing Required and Expected Gas-Temperature Curves on 1000 F—1000 F Reheat Unit Em- ploying Adjustable Tilting Burners

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[2] Numbers in parentheses refer to the Bibliography at the end of the aper.

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Note: Statements and opinions advanced in papers are to be nderstood as individual expressions of their authors and not those of he Society. Paper No. 48—A-106.

the reheater, as required to maintain a constant 1000 F reheat temperature from 60 to 100 per cent of design capacity, and curve *C-D* is the corresponding gas temperature leaving the furnace, required for 1000 F primary-steam temperature at design capacity. The shape or flatness of these curves is affected not only by the steam-temperature variation for which they are plotted, but also by the degree of variation in feedwater temperature and excess air. The position of these gas-temperature curves with respect to the vertical scale is affected by the design difference between the gas temperature leaving and steam temperature entering the cold-end section of the superheater. A low-temperature difference results in a lower required gas temperature leaving the furnace but also results in greater heating surface in the superheater and reheater. Therefore a higher gas temperature leaving the furnace permits the use of a smaller and less expensive superheater.

The curve of actual gas temperature leaving a furnace, fired in a conventional manner, is much steeper than the curve of gas temperature required to maintain constant steam temperature. Therefore, in Fig. 1, if the temperature indicated at *E* is required to give the specified steam temperatures at 60 per cent of capacity, and fixed turbulent burners are used, the gas temperature leaving the furnace at 100 per cent capacity will exceed the required gas temperature *D*. In the interest of cost economy of furnace and superheater, it is desirable to select the furnace so that point *D* is below the temperature which will cause troublesome ash deposits. However, this procedure cannot be followed with fixed burners even of the tangential type (which have a flatter gas-temperature characteristic than the turbulent type), except at a sacrifice in steam temperature at reduced load.

However, Fig. 1 illustrates a solution to the problem through the use of adjustable (tilting) burners with the furnace designed as previously mentioned. It will be seen that the maximum gas temperature leaving the furnace need not exceed the values indicated by the line *E-D*, because the range of control of furnace gas temperature will be as indicated by the dash lines *X*, *Y*, and *Z*, which are drawn for three different positions of the tilting burners.

A more detailed discussion of the use of tilting burners for furnace gas-temperature and steam-temperature control will be found in references (14, 15, 17, 18). Another series of papers (16), published by this Society, reports quantitative data on variation in furnace heat absorption and furnace gas temperature, based upon actual tests conducted on a large steam-generating unit employing this method of firing. In these tests a maximum gas-temperature variation was obtained at the furnace outlet at constant boiler load by the use of the adjustable burners, which greatly exceeds that used in the study illustrated in Fig. 1. If that degree of gas-temperature control had been used, an even greater range of load at constant steam temperature could have been predicted.

Another series of tests, reported in reference (18), are illustrated in Fig. 2. The solid line *A* is the gas temperature required to hold constant steam temperature over a 5 to 1 load range on a nonreheat unit delivering 900 F steam. The shape of the curve was affected by variation in excess air and feedwater temperature with variation in load, and the curve was plotted from data obtained with the supplementary steam-temperature control (by-pass dampers) closed. The dash lines *B*, *C*, and *D*, are gas-temperature curves with burners tilted up 25 deg, horizontal, and down 25 deg, respectively. Superposition of the burner-tilt curves on the required temperature curve illustrates how a 5 to 1 operating range on this unit can be obtained with constant steam temperature, using burner tilt only for steam-temperature control.

The heart of the steam-temperature control system of these modern reheat boilers is the burner illustrated in Fig. 3, one of which is located in each of the four furnace corners. These

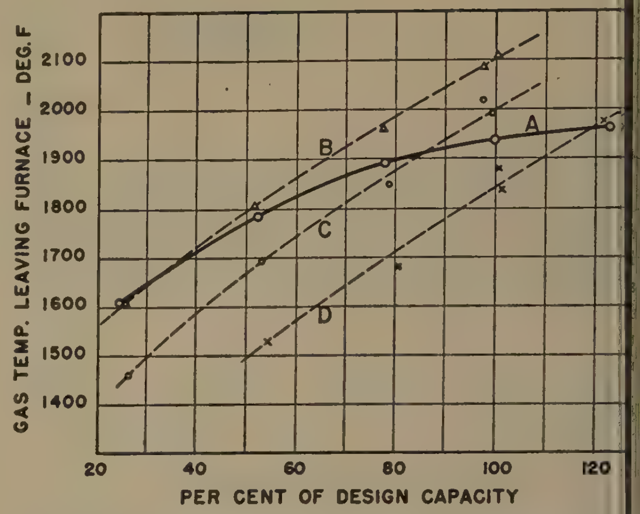


Fig. 2 Test Results on Nonreheat 900 F Unit Showing L
Range Obtained With Tilting-Burner Superheat Cont

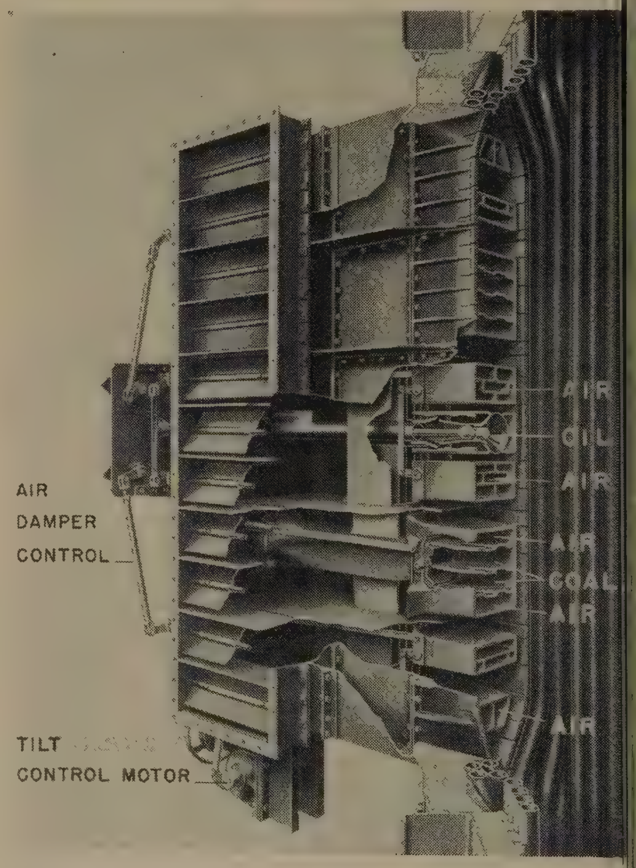


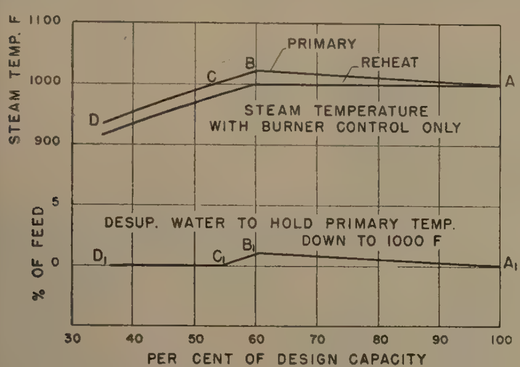
Fig. 3 Typical Burner for Corner Firing, Illustrati
Pivoted Fuel and Air Nozzle Tips, Tilting Mechanism, A
Air-Damper Operating Mechanism

burners may be designed for firing coal, gas, or oil, separately simultaneously, and all the fuel-nozzle tips and air-guide va are moved or tilted simultaneously by a motor-operated mech nism actuated by impulses originating at a steam-temperatu thermocouple. An air-flow element in the control system serves anticipate a steam-temperature change when a load change is perienced, thereby providing a more nearly constant steam te perature than would exist if the thermocouple only were used

ɔntrol purposes. In addition to operating the burner tilt ıechanism, the control system may include supplementary con·ol means for operating a by-pass damper or water valves for a ɔray desuperheater.

By proportioning and locating the reheater properly, the final eheat and superheat temperatures are both controlled by burner lt, but the tilt angle is automatically actuated by changes in re-eat temperature. The temperature of primary steam may equire supplementary control by spray desuperheating.

Fig. 4 shows typical curves of steam-temperature and desuper-eating-water requirements. The upper curves illustrate steam-emperature variation over a wide load range when burner tilt is ctuated by the reheated-steam temperature to hold 1000 F over ontrol range. It will be noted that when the primary superheater also proportioned to give 1000 F steam temperature at design



IG. 4 Application Analysis Showing Expected Primary-team and Reheat Temperatures With Burner Control Only, nd Amount of Desuperheating Water Required for Supplementary Control

apacity, the primary-steam temperature will increase gradually long the line A-B as load is reduced to the control point, and then ecrease along the line B-D. Therefore, when operating in ac-ordance with design conditions, no supplementary control will e required by the reheater. However, with the controls set to old the primary-steam temperature also at 1000 F, a small mount of desuperheating water automatically will be supplied to ne primary-steam circuit. The amount required is indicated by ne line A_1-B_1-C_1-D_1, and this water is introduced between the two ctions of the superheater so that the average outlet steam tem-erature will not exceed the design value. A desuperheater of the ɔray type is also provided at the reheater inlet to supplement urner-tilt control, if and when needed, to compensate for large anges in feedwater temperature or other unusual operating con-tions.

A steam-atomized water spray is more universally adaptable to e on the primary superheater because some plants operate with ery low excess pressure on the boiler feed pumps, and there is fficient pressure drop across the low-temperature section of the perheater to provide the necessary differential steam pressure r steam-atomized water sprays which require much less dif-rential water pressure. Mechanical atomization is used for the rays at the reheater inlet and may also be used for the primary perheater in plants where sufficient excess water pressure is ailable at the feed pumps. In both desuperheaters the inlet eam is sufficiently superheated so as not to be reduced to satura-on by the small amount of water involved.

Steam-pressure drop between the high-pressure turbine exhaust d the low-pressure turbine inlet must be kept to a minimum in der to obtain maximum efficiency with the reheat cycle. How-er, a reasonably large pressure drop of 20–25 psi must be taken across the reheater proper, to insure adequate steam distribution and protection against overheating in individual circuits. Steam piping to and from the reheater therefore must be designed conservatively to reduce line losses.

Since the gas temperature entering the superheater may be close to the fusion temperature of the ash it is customary to use wide transverse tube spacing near the furnace exit and a much closer spacing in the gas-outlet section. Materials of course must be satisfactory for the temperature and stress encountered, and most installations have as many as four different alloy steels in addition to the carbon steel used at the steam-inlet end of the superheater and reheater. The tube diameter and circuit length must be proportioned to meet the allowable pressure drop, and the circuit arrangement must be chosen to insure an efficient temperature difference between gas and superheater surface in all parts of the unit in order to economize on heating surface.

The almost complete absence of convection steam-generating surface will be noted in all modern reheat-boiler units.

Typical Installations

One of the largest high-pressure topping installations employing an integral convection reheater is that at Montaup Electric Company where the boiler is designed for 2000 psi with primary-steam

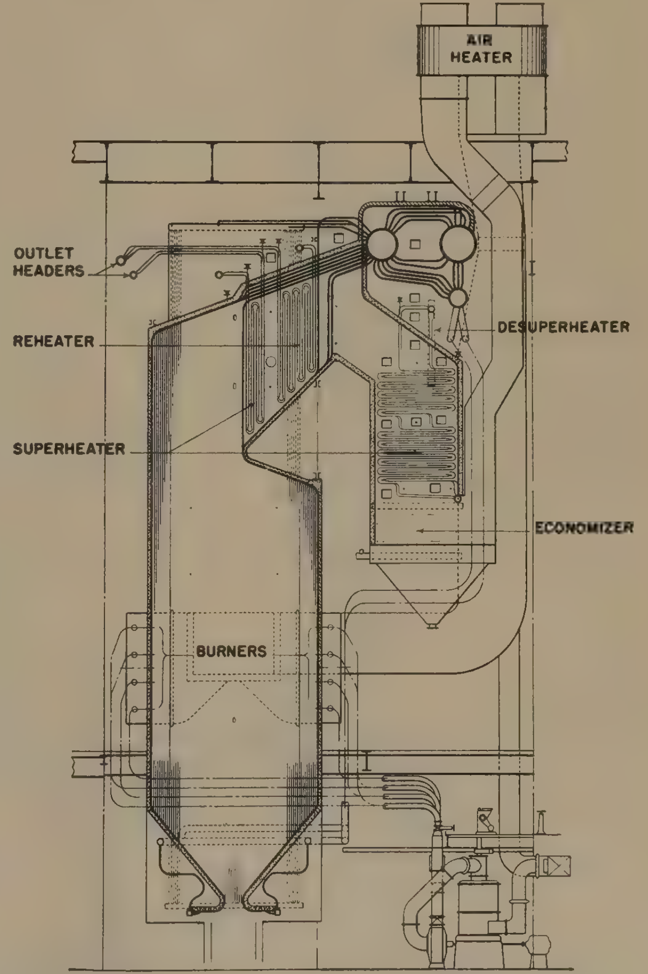


Fig. 5 Reheat Boiler, Coal-Fired, Water-Cooled Furnace and Hopper Bottom

(Design pressure 1650 psi; maximum capacity 670,000 lb per hr at 1000 F and 1492 psi; reheat flow 585,000 lb per hr at 1000 F and 406 psi. Controlled superheat and reheat with tilting tangential burners and spray desuperheaters.)

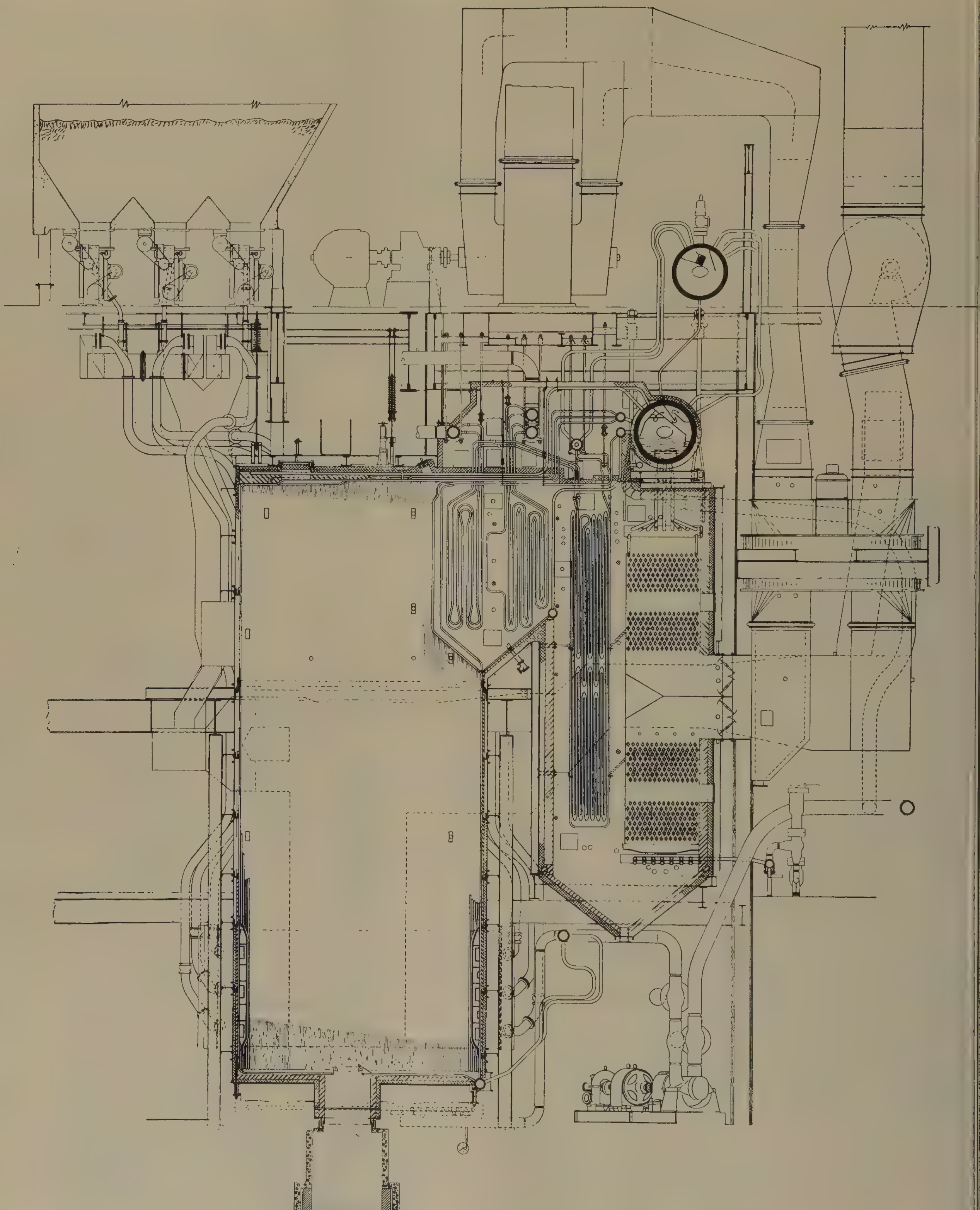


Fig. 6 Forced-Circulation Reheat Boiler, Coal or Oil-Fired, Water-Cooled Slag-Bottom Furnace
Design pressure 2000 psi; maximum capacity 650,000 lb per hr at 1825 psi and 960 F; reheat flow 567,000 lb per hr at 370 psi and 765 F. Super-heat controlled by use of by-pass dampers, and reheat by spray desuperheater at inlet.)

temperature of 960 F and reheat of 765 F. This unit dates back to 1940, and much of the experience gained thereon has been applied in projecting newer designs. The unit is illustrated in Fig. 6 where it will be noted the reheater is positioned the same as on more recent units. Montaup has a slagging bottom and was designed before the advent of tilting burners, but superheat control is obtained with a novel by-pass arrangement which has been duplicated on many other units. Reheat-steam temperature is controlled by a spray desuperheater at the reheater inlet.

Fig. 5 is typical of twelve dry-bottom units now in process of design or construction by the author's company for seven public-utility companies. They all employ the same division and general arrangement of superheating sections with flexible outlet terminals on both superheater and reheater to permit a simplified design of high-temperature piping to the turbine. The position of the burners is indicated, and it will be noted that a large portion of the water-cooled walls is located below the burners, in order to take advantage of downward burner tilt when necessary to counteract the effects of poor coal or dirty furnace walls, or to compensate for the effect of large changes in feedwater temperature. There is a striking similarity to nonreheat units currently offered by the author's company, the most noticeable difference being that more space is occupied by the superheater and less by the economizer.

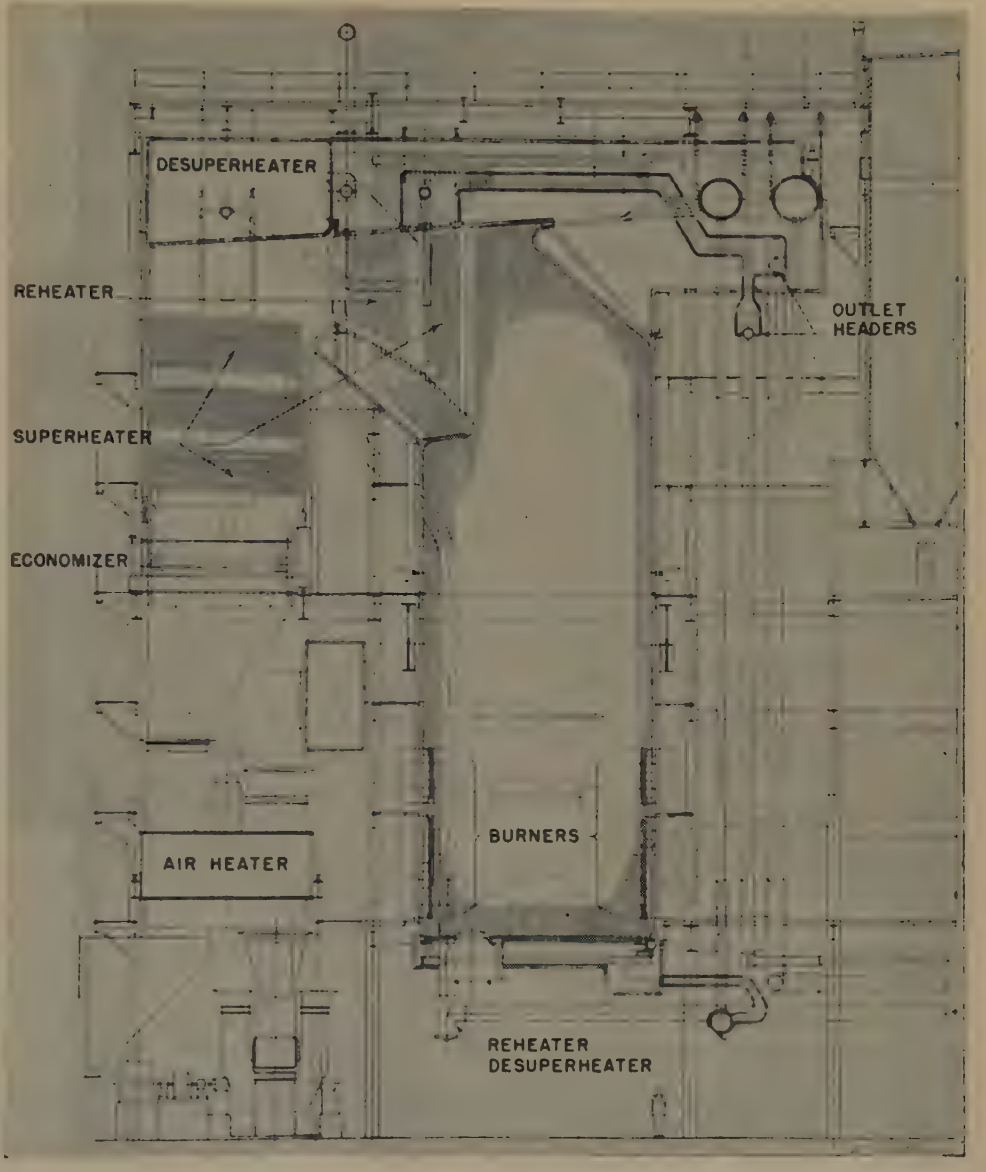


Fig. 7 Reheat Boiler, Coal or Oil-Fired, Water-Cooled-Slag-Bottom Furnace
(Design pressure 1770 psi; maximum capacity 950,000 lb per hr at 1050 F and 1578 psi; reheat flow 830,000 lb per hr at 1000 F and 380 psi. Controlled superheat and reheat with spray desuperheaters supplemented by tilting tangential burners.)

Fig. 7 shows the preliminary drawing for a unit now in process of manufacture for a large utility company in New Jersey. This is of the wet-bottom continuous-slag-tap design and it will be noted that the burners are divided into two groups, the lower to insure continuous removal of slag, and the upper to assist in steam-temperature control through burner tilt. Primary superheat control is provided through spray desuperheaters at the reheater inlet and also between the front and rear sections of the primary superheater, both of which are under automatic control. The burner-tilt control probably will be remote manual.

In all these units the high-temperature section of the primary superheater is placed nearest the furnace outlet, and this section has the tubes spaced on wide centers to reduce ash accumulation when using low-fusion-ash fuels. Behind this section is the low-pressure reheater, and beyond that the low-temperature section of the primary superheater. These sections are placed in series with respect to gas flow, and there are no longitudinal partitions or baffles.

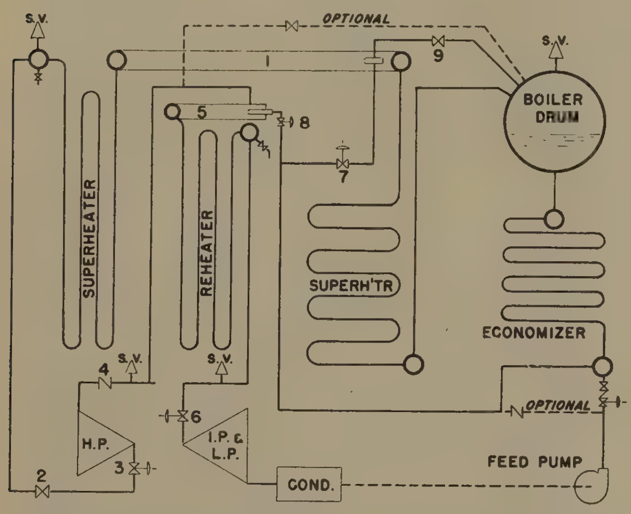
The furnaces of these typical recent designs, when selected to give a gas temperature that will not cause troublesome ash deposits at the furnace exit, with tilting tangential burners, will have a maximum gross combustion rate of about 18,000 Btu per hr per cu ft and a net heat-release rate of about 115,000 Btu per hr per sq ft with complete water cooling, using bare wall tubes with tangent spacing.

Operation

The use of a single large boiler, turbine, and reheater all connected in series without cross-connection at either the high- or low-pressure level simplifies design and operation and reduces the capital investment compared to small cross-connected units.

When the unit is installed in an existing low-pressure station, a steam source may be available to protect the reheater during starting up, but many of the recently purchased reheat units are

going in new stations. Means of self-protection within the unit may be provided as illustrated in Fig. 8. With this arrangement, maintenance of steam flow through the reheat section is accomplished as readily as through the sections of the primary superheater. At extremely reduced boiler loads or when the turbine is on spinning reserve, steam is always passing through the reheater section. To all intents and purposes these units constitute, therefore, essentially a conventional standard high-capacity utility-type steam generator with another superheater section interposed, and the operation is no more difficult and requires no more personnel.



1 Primary steam desuperheater
2 Emergency stop valve
3 Governor-operated valves
4 Optional reheater stop or intercept valve
5 Reheat desuperheater
6 Reheater intercept valve
7 Water-control valve, primary steam desuperheater
8 Water-control valve, reheat desuperheater
9 Steam stop valve, primary steam desuperheater
10 Optional steam line for reheater protection

Fig. 8 Schematic Diagram Showing Typical Interconnections to Turbines and Desuperheaters on Reheat Boiler

Prime movers for connection to reheat boilers are furnished by several companies, each of which may have a different method of protecting the turbine during emergency periods resulting from sudden loss of load. Means for protection of the reheater and turbine, therefore, must be worked out in conjunction with the turbine manufacturer for each individual installation. Several methods of accomplishing satisfactory protection are discussed in considerable detail in other papers of this symposium (19), (20), (21), (22), (23), (24).

Automatic control devices have been developed to such an extent that routine control and protection within reasonable limits can be accomplished without much attention from the operator.

Obviously, a greater capital expenditure can be justified when a high load factor is used, and these reheat units will normally be base-loaded whenever possible, in order to realize the approximately 5 per cent lower fuel consumption. However, it is a distinct advantage to have a unit capable of maintaining the design maximum steam temperature from both primary superheater and reheater at reduced output, and the method of steam-temperature control used on these recently designed units is admirably suited to accomplish this result with automatic control.

Conclusion

It may be stated that when all superheating and reheating can be accomplished with convection surface, these modern reheat units, as designed by the author's company, are not much different in appearance from large nonreheat units, since the reheate is simply placed between the primary and secondary superheate The secret of this simplicity lies in the ability to design a unit wit controlled gas temperature leaving the furnace through the use o tilting burners.

Acknowledgment

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20 "Developments in Resuperheating in Steam Power Plants," by E. E. Harris and A. O. White, Trans. ASME, vol. 71, 1949, pp. 685–691.

21 "Steam Turbines for Resuperheat Cycle," by E. E. Parker, Trans. ASME, vol. 71, 1949, pp. 693–700.

22 "Reheating in Steam Turbines," by R. L. Reynolds, Trans. ASME, vol. 71, 1949, pp. 701–706.

23 "Steam Generating Equipment for Resuperheating Cycles," by Martin Frisch, Trans. ASME, vol. 71, 1949, pp. 707–717.

24 "High-Pressure Boilers With Reheaters," by W. H. Rowand, A. E. Raynor, and F. X. Gilg, Trans. ASME, vol. 71, 1949, pp. 715–727.

Symposium on the Reheat Cycle—Discussion

At the Annual Meeting of The American Society of Mechanical Engineers, held in New York, N. Y., November 29–December 3, 1948, a series of seven papers on various aspects of the reheat cycle were presented under the auspices of the Power Division. The discussion which follows relates to one or more of these papers and so is published as a composite discussion of all, reference, however, being made to individual papers wherever necessary.

The seven papers and their authors are listed in order of presentation; the numbers appearing in parentheses throughout the discussion will refer to the corresponding numbers in the list. Pages given refer to this issue of the Transactions, in which they appear in the same order.

1 "Operating Experiences in Connection With Regenerative Reheat Turbine Installations," by C. A. Robertson, pp. 673–683.

2 "Developments in Resuperheating in Steam Power Plants," by E. E. Harris and A. O. White, pp. 685–691.

3 "Steam Turbines for Resuperheat Cycle," by E. E. Parker, pp. 693–700.

4 "Reheating in Steam Turbines," by R. L. Reynolds, pp. 701–706.

5 "Steam-Generating Equipment for Resuperheating Cycles," by Martin Frisch, pp. 707–717.

6 "High-Pressure Boilers With Reheaters," by W. H. Rowand, A. E. Raynor, and F. X. Gilg, pp. 719–727.

7 "Modern Reheat Boilers," by W. S. Patterson, pp. 729–734.

The Symposium was conducted by Philip Sporn, Fellow ASME, and president of the American Gas and Electric Service Corporation, New York, N. Y. At the opening of the session and later through the presentation of papers, Mr. Sporn addressed the meeting on salient points of the reheat cycle and its applications. His remarks appear in the following as they pertain to the subject matter of the symposium record.

Comments by Philip Sporn, Symposium Chairman

The interest which the reheat cycle has finally evoked after 25 years of demonstrated initial and successful installation is reflected not only by the symposium, which is reported in this issue of the Transactions, but by the following steam-turbine units on order for delivery in the 3-year period 1949, 1950, and 1951:

	No. of units	Kilowatt rating
General Electric	22	1,920,000
Westinghouse	6	492,000
Allis-Chalmers	3	240,000
	31	2,652,000

Certainly from the standpoint of both the number of units and the total kilowatt rating of these units, this is a very striking illustration of the coming into its own of the reheat cycle.

I have inquired into the cause of this much-delayed, but gratifying emergence of interest in a tool which was so promising from its very beginning, and have concluded that the following causes are responsible:

Development in Cost of Fuels. Not only have costs risen sharply at the source of supply, but transportation costs have risen greatly and threaten to rise further. From this has come a realization that for some period ahead, cost trends are not likely to be reversed sharply, if at all.

Design Developments Both in Boilers and Turbines. This is best illustrated by the developments of the tandem unit described by Mr. Parker, (3) and by the boiler developments discussed by Messrs. Frisch (5), Patterson (7), and Rowand and associates (6). In essence all of these have shown ways to reduce the differential in cost between reheat and nonreheat setups.

Operating Experience. For reasons which are really hard to explain, reheat, independently of the economic justification it has always had to demonstrate, has had to overcome an obstacle of no mean proportion in its supposed complication and possibly even operating hazard. The writer cannot forget that the steam engineers working on the original reheat Philo units 1 and 2 in 1923, were deeply concerned over the effect on safety of the turbine in case of loss of load, owing to the greater steam inertia that the unit seemed to possess from loss of reheat. They asked him, therefore, to design a load that would be thrown automatically on the machine in case of such an emergency, and he did. Fortunately, good sense prevailed, and the combination of resistor and low-tension switch to accomplish this loading was never installed. Nor has a need for it developed on this or any of the other ten reheat units built on the American Gas and Electric Company system and now operating. Incidentally, six additional reheat units, with a total capacity of 900,000 kw, are now in the process of installation at three plants on the American Gas and Electric Company system.

With increased use of reheat will come not only added confidence in its efficacy as an important means for improving efficiency of large-scale generation of electric energy, but design improvements that will further help the process along.

It is particularly timely to re-examine now all the basic thermodynamic, economic, design, and operating factors of the generation phase of the reheat cycle.

Reheat boilers can be made very special and complex. We have had experience with a variety of designs, including live-steam reheaters, separately fired reheaters, and a number of types of integral reheaters. From all this there seems to be emerging a simplified functional integral reheater unit which is so well-adapted to the growing practice of single boiler-turbine installations. The reheat piping under such conditions is a relatively direct and simple system and this is a material aid in keeping the incremental cost of reheat under control.

Papers (5, 6, 7) represent the very best in the way of real engineering data on this timely subject. The authors are all directly engaged in the actual development and design of reheat-type generating units. They have had the benefit of the available past experience and present the most thorough and up-to-date analysis of the subject.

The artist's portrayal of the Philip Sporn Plant, as shown in Fig. 9 of the paper by Messrs. Rowand, Raynor, and Gilg (6) is already obsolete. The two stacks representing the initial two units have already grown to four such units under construction—a program which will give us 600,000 kw net capacity in this plant by 1951. It may be of interest to note that this plant is projected for six units and that when completed it is expected to generate at a rate of 6,000,000,000 kwhr per year for the dominant part of its life.

Since the high-pressure high-temperature steam cycle is bound to play the most prominent part in the expansion of the country's power-generating facilities during the next 10 to 15 years, at least, the writer cannot help but feel that we must and should be searching for the next higher level of performance, which appears logically to be at higher steam temperatures—with or without reheat. In the writer's judgment it will be more likely with reheat.

Discussion

W. E. Caldwell.[1] From the wealth of information presented in these excellent papers two important benefits of resuperheating stand out.

1 Improvement in economy of power production of about 4.4 per cent, or roughly, 400 Btu per kwhr.

2 Reduction in exhaust moisture of about 50 per cent with resulting decrease in turbine maintenance.

The authors have indicated the gain in efficiency is due to two causes:

1 Increased efficiency in lower turbine stages due to moisture reduction.

2 Thermodynamic gain due to increase in temperature in the low-pressure unit.

The major part of the gain is due to the increased efficiency which accompanies moisture reduction below the dew point and principally in the subatmospheric stages.

Fig. 3 of the Harris and White paper (2) indicates that about one half the usual exhaust moisture still remains with conventional resuperheat to present limits. The loss resulting from this moisture also warrants recovery if it can be realized within the limits of economic justification.

In a nonreheat turbine, the moisture increases about $1^1/_2$ per cent per stage in the last eight stages below the dew point, and with simple design modifications this water could be evaporated by incorporating a relatively small amount of heated surface in each wet stage. Measurements from existing turbines indicate the nozzle surface in the wet stages is sufficient to evaporate this moisture if the nozzle blades can be heated a moderate amount above the corresponding wet-stage temperature. By the use of hollow nozzle partitions of suitable material supplied with steam, in the hollow space, bled from upper stages, it is possible to evaporate the moisture as it is formed. By evaporating the moisture with waste heat and without pressure loss the maj portion of the gain obtainable by external resuperheating may realized. The measured nozzle surface and calculated ble pressures required for the evaporative duty were contained a paper.[2]

A simple means of accomplishing this evaporation within th turbine is shown in Fig. 1, herewith. For illustration, two h stages are shown but ordinarily more would be justified by ec nomic considerations. Steam bled from upstream stages ente the top of the diaphragm, through a cored passage, flows throug the inside of the hollow blades into the center segment, then in to the lower blades to the bottom drain. The condensate forme in the hollow blades would be discharged through a drain in th shell at a temperature approaching that of the bleed steam use for the heating medium. This condensate may be returned t the feed system by draining it into the bleed-heater condensa system at the proper stage.

This subsaturation method of reheating requires no extern apparatus other than drain connections, introduces no pressu loss in the steam path, and has the same self-regulating characte istic as regenerative feedwater heaters. For the heating mediu waste heat may be utilized which otherwise would pass to th condenser. To provide the necessary temperature differenc for the required heat transfer, a loss of about 60 Btu (of availa ble heat) per lb of heating steam must be taken. By sacrific ing this 60 Btu (adiabatic heat drop), 900 Btu (enthalpy of evapo ration) otherwise rejected to the condenser, is made availabl for evaporating moisture in the turbine. By this means th latent heat from 10 per cent of the steam may be used to evapo rate the moisture in the remaining 90 per cent, thus increasin the efficiency and heat available to the stages below the dew poin

The condensate from the heating nozzles may be returned t the feedwater system with insignificant loss due to temperatur degradation. The gain in economy is not impaired by increas

[1] Staff Engineer, Consolidated Edison Company of New York, Inc., New York, N. Y. Mem. ASME.

[2] "Subsaturation Reheat Cycle," by W. E. Caldwell, presented a the Annual Meeting, Atlantic City, N. J., December 1–5, 1947, o The American Society of Mechanical Engineers, Paper 47—A 110.

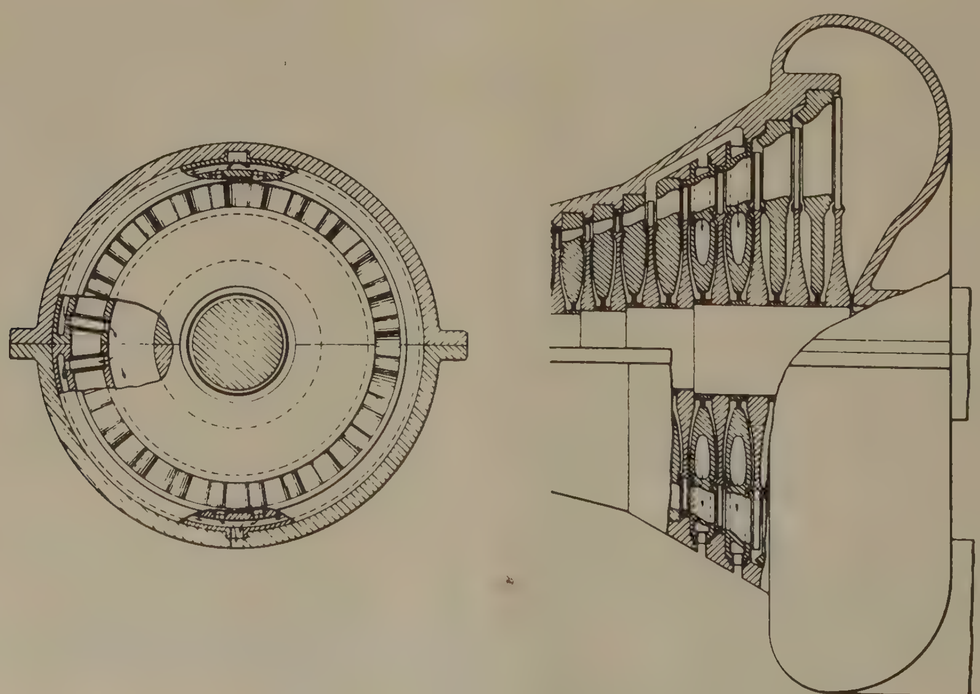

Fig. 1

in number of feed-heating stages in the manner indicated in the Reynolds paper (4) in Fig. 5, applying to the conventional resuperheat cycle.

Subsaturation reheating offers attractive possibilities for economic gain either with or without external resuperheating.

By evaporating the moisture within the turbine there is a possibility of realizing over 60 per cent of the increased economy attainable with conventional resuperheating for about 10 per cent of the increased investment cost.

A. G. Christie.[3] The group of papers dealing with reheating steam turbines present much information of interest to engineers who have to do with power generation.

Many are familiar with the earlier installations of reheating turbines. As outlined in the several papers, these early turbines have developed creditable operating records. Few will question the adaptability of such turbines to operate satisfactorily under certain load conditions. In general, such units are given base loads during the early years of operation. But more economical later plant additions to a system will force these older machines into peak-load operation. This load condition requires frequent starts and stops and often large load swings. The question arises: "How well will reheat units operate under such load conditions?"

It is pointed out in one of the papers that light-load operation may lead to overheating of the exhaust section unless provision is made for desuperheating the steam from the reheater. This will offset some of the thermal gains at light loads from the use of reheat.

On sudden loss of load it is necessary to stop the flow of steam from the reheater to the low-pressure section of the turbine to prevent overspeed of the unit. Two alternative methods may be used. The steam in the reheater may be allowed to discharge to the atmosphere through a relief valve. However, as this requires added make-up, it would not be an acceptable method in a modern station. The second method is to discharge this steam to the condenser through an unloading valve. This steam would be highly superheated, and the desirability of suddenly discharging a large volume of such steam into a condenser shell and up around the last wheels of the turbine may be open to question. A more serious difficulty would be the loss of vacuum which Mr. Robertson (1) says would amount to 6 or 7 in. of mercury. It would take an appreciable interval to recover the full vacuum after such an occurrence.

The thermal gains from reheat of 4 to 6 per cent are important factors in favor of reheat, particularly where coal is high in price. Against this saving one must balance the increased cost of the whole reheating plant over a nonreheat plant. A further consideration is the increased complexity of the reheat plant in both design and operation.

Reheating units will be installed where system base loads can be carried and fuel costs are high. Where loads are variable, the simpler nonreheat cycle may be selected.

M. K. Drewry.[4] Mr. Parker (3) does a good service to the power industry by calling attention to the possibility of unduly high exhaust temperatures accompanying reheat at very low loads.

Equilibrium exhaust temperature of 500 F is estimated for Port Washington turbine No. 3, under conditions of no load and normal speed after sudden full-load loss, based upon a rapidly rising 232 F exhaust temperature, observed when testing the safety governor 10-5-48. Reheater-outlet temperature was only 600 F, and exhaust pressure only 0.5 in. abs, but the 232 F would have been exceeded considerably, because it was rising steadily, had the speed not been reduced promptly to well below normal.

Higher exhaust-blade speed, not necessarily reheat alone, has contributed to this high exhaust-temperature condition. Port Washington turbine No. 2, with 18 per cent less maximum blade-tip speed (929 fps, versus 1135 of PW 3) operated for 24 hr at about 1000 kw for initial electrical tests without superheat appearing in its exhaust. Its superheat and reheat temperatures are approximately 50 F below those of PW 3.

Approximate estimates of exhaust temperature can be made by determining work done on the steam by acceleration to the mean blade speed of every stage after the point where blade velocity exceeds steam velocity. The "square-law" that applies causes superheating of exhaust steam to increase quadratically with blade speed.

Low no-load losses, resulting from hydrogen generator cooling, less bearing friction, etc., have so acted to increase no-load exhaust temperatures that they apparently can become evident on modern high-temperature nonreheat units upon full-load loss if normal speed is held too long before reloading.

The following defenses against exceeding 250 F maximum allowable exhaust temperature upon full-load loss of Port Washington turbine No. 3 are, in effect, though not totally tried by experience:

1 Minimum practical steam-inlet temperatures.
2 Maximum possible vacuum.
3 Only one inlet valve open.
4 No throttling of intercepting valves.
5 No by-passing of hot steam into low-pressure blading.
6 Minimum practical time at or near rated speed before synchronizing and loading adequately.

Maximum blading efficiency obviously is necessary to obtain minimum heat content of exhaust steam, which suggested precautions 3, 4, and 5.

Heat capacity of the low-pressure blading is relied upon to afford enough time to resynchronize upon full-load loss without exceeding safe exhaust temperatures. Contrarily, heat capacity of the steam piping acts unfavorably by sustaining high steam-inlet temperatures.

Experience with desuperheating in the crossover pipe of PW turbine No. 2 on an occasion of blade loss in 1945, indicates that this defense against high exhaust temperature requires careful design and careful operation. Water hammer occurred upon trial use of spray nozzles, and temperature reductions were slight. Other defense methods appear more desirable, but possibly the moderate use of all may be best.

Lesser moisture in the turbine exhaust, higher blade speeds, and lesser no-load losses are conducive of higher efficiency, but like most other efforts for lower production costs, they make the power-generation technique increasingly involved. However, the industry probably will cope satisfactorily with them as it has with the many other similar past improvements.

Mr. Frisch (5) presents a wealth of information on an involved subject in which the power industry has become vitally interested. He evaluates in directly usable figures the net results of many variables, affording valuable conclusions which many engineers do not have time to derive.

Acceptance of some of his conclusions, which probably will be surprising to many, may be eased by the truth that at last superheating duty is exceeding that of evaporating duty. The concept that superheaters (including reheaters, of course) are no longer auxiliaries of the boiler, but that "tables have turned,"

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[4] Assistant Chief Engineer of Power Plants, Wisconsin Electric Power Company, Milwaukee, Wis. Mem. ASME.

may allow easier acceptance of the author's conclusions 2 and 3. Higher pressures, with markedly higher specific heats, elevate superheating duty more than comparison of steam temperatures alone would indicate. Likewise, they reduce evaporating duty importantly, with the result that some stations in operation and in prospect use more heat for superheating than for evaporating.

Reheat and radiant superheaters have been used concurrently and consistently by the writer's company since initial starting of Lakeside boiler 17 in 1926. Sixteen earlier Lakeside boiler units have used radiant superheaters since 1925, resulting in a grand total of successful experience of "radiant-superheater years" equaling 500. Thus:

			Radiant-superheater years to date
Lakeside boiler room no. 1	8 boilers	1925 starting	184
Lakeside boiler room no. 2	8 boilers	1924 starting	192
Lakeside boiler room no. 3	5 radiants	1929 avg stdg.	95
Port Washington no. 1	2 radiants	1935 starting	26
Port Washington no. 2	2 radiants	1943 starting	10
		Total	507

These data, and the fact that newer Port Washington units 3, 4, and 5, are using radiants like Lakeside's and Port 1 and 2's, is seemingly abundant support of Mr. Frisch's statement, ".. in ... modern units difficulties attributable to the use of radiant superheaters are negligible."

That reheat plants, considered in their entirety and compared with others of similar coal versatility, are essentially identical in cost of nonreheat plants has been the experience of the writer's company, in agreement with Mr. Frisch's conclusion 1.

Careful study of reheat-turbine overspeed safety seems seriously needed. Mr. Robertson's paper (1) does very well to focus attention on this vital subject.

While reheat possibly doubles chances of destructive overspeed, experience indicates that nonreheat turbines are not of desirably high safety. The writer's company has experienced failure of a throttle trip linkage while inlet-valve leakage was steadily taking the unit to high overspeed. Another of its units could have suddenly reached destructive overspeed when its safety governor was tested many times by use of an improper testing device which suddenly opened all inlet valves wide. Another company reported simultaneous failure to close of throttle and inlet valves, and several experiences of carry-over holding both these vital valves open coincidentally have been learned. Destruction of a newly installed industrial turbine, due to entry of steam through a bleeder connection, represents the extra hazard of reheat.

Reheat-system steam, normally capable of causing about 30 per cent overspeed, is not noteworthy because of great capacity to cause overspeed but because of its considerable opportunity to cause overspeed. Whereas standard practice establishes two lines of defense at the main turbine inlet, it usually provides only one line of defense at the reheat stage. Assuming individual valve reliability to be 99.9 per cent of the times the valve needs to act to prevent overspeed, probability of overspeed, chargeable to reheat, is thus 1000 times that of simultaneous failure of the throttle, and a single inlet-valve design. Whether two lines of defense at the reheat zone are desirable seems to merit careful attention.

That 30 per cent overspeed is reasonably safe may be said of some turbines, and claims may be made that reheat valves serve only to meet the usual 110 per cent speed control upon rated load loss. If shop tests of the completed parts are run at 130 per cent speed, under normal operating conditions, and if there is positively no deterioration of strength after a lifetime of service, then the foregoing contention has a partial basis of logic. It omits allowances for maximum capability loads, for minor leakage of the many valves which may contribute importantly toward the very small flow rate required to overspeed a modern turbine, and for the many exigencies that have a habit of appearing eventually at a crucial time.

Two reheat-to-condenser unloading valves have the important virtue that two adequate lines of defense are afforded. They cause no continual pressure drop, which readily may capitalize into values exceeding the total cost of intercepting valves. Each need not be full size, or adequate alone to limit speed to 110 per cent, and their cost appears not greatly more than that of the reheat safety valves they displace. As Mr. Robertson states, "dump" valves are less likely to foul from foreign objects, and they have the inherent vital advantage, especially needed on 3600-rpm units because of their low inertia, that their maximum effectiveness occurs at the beginning of their travel; not near the end of travel as with intercepting valves. The steam they admit to the condenser exerts a powerful braking effect on the large exhaust wheels, the amount of which can be evaluated quite accurately.

Representing much actual practice, Mr. Robertson's paper (1) is commended to all engineers interested in reheat applications.

J. N. Ewart.[5] The paper by Mr. Parker (3), describing the development by his company of the steam turbine for the resuperheat cycle, is of considerable interest to central-station designers and operators. The paper not only describes the merits of this cycle from an economy standpoint but also points out quite frankly that there are certain problems which, though not insurmountable, must be reckoned with.

A serious disadvantage of the application of the reheat cycle during the late 1920's and early 1930's came about because of the complexity in both arrangement and operation of the so-called header system of grouping two or more boilers to a common header supplying one or more turbines. This practice was used quite generally at that time because the boiler was not considered to be as reliable as the turbine, and therefore one or more spare boilers were provided in the design. Hence it is not surprising that, with the jump in pressures and temperatures which took place during the middle 1930's, many designers chose that avenue in their quest for better heat rates rather than undertake the many problems then associated with reheat.

Steam units of the Niagara Hudson System must be designed to firm up a substantial amount of stream-flow hydrocapacity. As such they must be capable of rapid load changes in the event of transmission-line interruptions and capable of full rated continuous output for long periods of minimum stream flow which might occur during either the summer or winter months. This requirement demands reliability and a high degree of flexibility in operation. To provide for these requirements and keep capital investment to a minimum, for the past 10 years we have utilized the so-called unit system, employing the single boiler - single turbine arrangement with no interconnection between units.

Successful operating experience with the unit system has made it possible to eliminate the serious objection which formerly was associated with the use of reheat. In fact, the use of reheat with the unit system lends itself so readily that it probably will not again be abandoned with still higher temperatures as was done during the jump in temperature of the last period.

The author (3) states that the resuperheat turbine is as flexible with respect to starting time and rate of load change as the nonreheat turbine. In order to take full advantage of this, the boiler and its reheater must be capable of similar flexibility and without sacrifice of the simplicity inherent in the "unit" system. The

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single boiler - single turbine arrangement introduces a problem of protecting the reheater against overheating while lighting the fires and bringing pressure up on the boiler prior to taking flow to the turbine. It is most desirable that the reheater be built to take this temperature until minimum pressure can be obtained to start the turbine and thus create flow. There appears to be no basic reason why a turbine designed for 1500 psi, for example, cannot be started at a boiler pressure as low as 50 psi. This enables cooling steam to be established in the reheater before excessive gas temperatures can occur.

The recent improvements in equipment for this cycle, enabling a reduction of capital investment as stated in this paper (3), is most heartening at a time when costs are rising continually for practically everything that goes into a power plant.

S. N. Fiala.[6] The development of the reheat cycle has moved along steadily with the development of the more common nonreheat cycle. During this growth period, it has maintained its advantage with a premium in thermal efficiency of some 5 per cent.

Members of the writer's company have been pleased to note the renewed interest, on the part of engineers and executives of the power industry, in the reheat cycle. For our part, the use and improvement of this cycle have moved steadily onward since the middle 1920's, when the first units were installed on the American Gas and Electric system. Through the years the use of reheat with its thermal advantages has played a major role in our building program as represented by Philo, Stanton, Twin Branch, and Deepwater stations. The capacities in the reheat cycle of these plants constitute nearly 25 per cent of the entire reheat capacity installed in the country; 591,500 kw out of 2,400,000 kw.

Our operating experience with reheat cycles has been characterized by the absence of inherent operational difficulties peculiar to the reheat cycle. Early installations involved a separate boiler for reheating, while, on our most recent unit, using the 2500-psi natural-circulation boiler at the Indiana & Michigan Electric Company's Twin Branch Plant, all primary steam-making and all reheating were accomplished in a single boiler. This high-pressure unit has given such reliable operation that over a period of 7 years, after a preliminary shakedown, its availability has been 95.2 per cent with a load factor of 87 per cent, and a heat rate close to 10,300 Btu per net kwhr.

From the standpoint of design, additional surface must be proportioned, and additional controls must be provided for the boiler. For economic reasons and to save space, we have found it desirable to include the reheat surface in the main boiler. This single boiler is then fired for capacity, with initial control for superheat temperature and secondary control for reheat temperature. It has been found that reheat surface should be kept in the convection section of the boiler so that the reheater surface may keep cool without venting, under conditions of no flow or of low steam flow. This convective location also keeps the reheat temperature secondary to the primary superheater, which may have some of its surface located in the radiant zone.

An investigation of the steam temperatures in the reheat turbine was recently made on the 2300-psi reheat unit at the Twin Branch Plant. This was conducted to study the problems involved in control of temperatures in low-pressure reheat turbines during start-ups when temperatures abnormally high for the load carried might be encountered. It is hoped that a paper on the findings of this test may be presented before the Society in the near future.

We have maintained a policy of engineering new units to the most advanced ideas in the science of power generation. Through the years this has shown us the advantages which can accrue from the use of reheat under suitable design conditions. Rising fuel costs have only intensified the need for economical production and therefore we are continuing the development and use of the reheat cycle. The American Gas and Electric Company's building program has six reheat units in progress, each with a capability of 150,000 kw. This will add 900,000 kw in reheat capacity to the system by 1951. These new units, to operate at 2000 psi, 1050 F initial steam temperature, and 1000 F reheat temperature, with a heat rate of 9300 Btu per net kwhr, the broad outlines of which were presented in a paper before this Society in 1947, will mark another milestone in the economical generation of steam-electric power.

[6] Mechanical Engineer, American Gas and Electric Service Corporation, New York, N. Y. Mem. ASME.

R. W. Hartwell.[7] During the present period of active investigation of resuperheating cycles, the information presented in the paper by Harris and White (2) should prove to be of considerable assistance to those interested in the economics of resuperheating. The authors are to be congratulated on a thorough analysis of the heat-rate gains inherent in the resuperheating cycle.

Only in the conclusion of the paper is reference made to the size of machines to which the heat-rate gain percentages apply. Can it be assumed that the same gains would be realized on machines of 60 mw capability as on machines of 125 mw capability?

That the increase in available energy in a reheat cycle reduces the steam-flow requirement for the same output is noted in the paper. As this reduction in throttle steam flow and flow to the condenser may be from 12–16 per cent for the 1250-psi 950-deg F cycle drawn in Fig. 3 of the paper, it seems that more emphasis might be placed upon this point, in view of its significant effect on size of steam generators and other equipment.

In selecting the reheat pressure, there are numerous practical questions which must be considered in addition to the thermodynamic gains and the desire to maintain economy at partial loads. One of these is the very high specific volume of 150–200 psi steam at temperatures of 950 to 1050 F, and the resultant need for large reheat piping from the boiler back to the turbine as well as the lead from the turbine to the boiler. Preliminary calculations indicate that the cost of the reheat piping might increase 40 to 50 per cent as the reheat pressure is reduced from 400 to 150 psi. Space limitations and flexibility problems also favor the choice of higher reheat pressures.

From an operating standpoint, the reduction in the moisture content of the steam as it passes through the last stages of the turbine represents an advantage of the reheat cycle that is worthy of additional comment. By reducing the moisture to 5–8 per cent, reheat minimizes the problem of turbine-blade erosion and the resultant maintenance expense. Over the life of a turbine, it is anticipated that this saving would be substantial.

Edwin H. Krieg.[8] The present renaissance of interest in power-plant economics, which has been spurred by higher fuel prices, quite properly has been directed toward the potentialities of the reheat cycle which offers gains equivalent to raising initial throttle temperatures by some 150 F. For equivalent heat rates, the reheat cycle is more conservative in many ways than the regenerative cycle, which would require about 150 F higher throttle temperature.

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[8] Consulting Engineer, Stone & Webster Engineering Corporation, Boston, Mass. Fellow ASME.

Reheat-cycle economics are sound. When designers wonder whether the incremental cost of the reheat cycle is justified, the question could well be rephrased: "How can the straight regenerative cycle be justified on units of 100,000 kw and over?" For units of around 125,000 kw, it is possible to justify reheat with fuel prices of 20 cents per million Btu and higher, and factors of 65 per cent or higher.

There need be no mystery about the economics of the reheat cycle. Mr. Frisch stated in his paper (5): "Steam generators for turbine generators employing resuperheating cycles cost little if any more and may, for a small steam-temperature control range, cost less than steam generators for nonresuperheating units for the same turbine capability and primary steam conditions." On many reheat studies, actual boiler prices have shown that reheat boilers cost only 1 to 3 per cent more than straight regenerative-cycle boilers for turbine generators of the same capability for 1300 psi. This differential may increase for higher pressures.

The cost differential between reheat and nonreheat turbines is easily obtained from the manufacturers, and this runs around \$1.50 to \$2 per kilowatt depending upon the size of the unit. At this point we do not check with the argument that reheat might be justified for sizes of turbine generators down to 20,000 kw. Beginning with the 60,000-kw size and smaller, the reheat cycle must compete with preferred standard turbines which cost some 12 per cent less than "old rating" turbines for the same name-plate rating without correction for "capability." In addition, the extra \$150,000 cost of a reheat 60,000 kw turbine is often just about enough to negate the economic advantages of reheat, and this is even more exaggerated for smaller turbine generators.

Restating this, in turbine-generator sizes where preferred standard units are available, reheating is usually difficult to justify because, first, a 60,000 kw 1250 psi 950 deg F nonstandard turbine costs about \$160,000 more than a preferred standard turbine, on a name-plate (not a capability) basis. In addition to this 12 per cent increase, a reheat unit would cost an additional \$150,000, making a total of approximately \$310,000. In turbine-generator sizes 80,000 kw and over, where no preferred standards exist, one does not have the penalty of the difference in cost between preferred standard and nonpreferred standard units; only the differential between reheat and nonreheat turbines exists.

There is some tendency to overestimate the cost of reheat piping by not crediting reheat with handling some 16 per cent less throttle steam. Condensate pumps, boiler feed pumps, heaters, condensate and boiler-feed piping are all smaller. Pumps take less power. Condensers handle some 13 per cent less steam and about $7^1/_2$ per cent less Btu. It must be that many estimates do not take all these details into account in pricing piping, valves, and auxiliaries. Where cooling water is limited, $7^1/_2$ per cent or more capacity may be obtained from a given river flow.

Then why wasn't the reheat cycle popular prior to the present sizable crop? Largely because confidence in the availability of large boilers was not general, especially when poorer grades of coal were used. The reheat cycle demands a single-boiler-turbine combination; this means single boilers with a capacity of 80,000 to 150,000 kw. Until a few years ago, boilers were forced to the point where availability was affected. Present-day boilers are, on the whole, more liberally designed; this means much improved availability and confidence in large single boilers.

Several of the papers presented share the same fault of not indicating clearly whether a cycle or a net plant heat gain is being discussed. When a gain of, say, 400 Btu is made on the cycle, this may increase to 490 Btu per kwhr on the net station heat rate when account is taken of boiler efficiency and auxilia[ry] power, and that makes a lot of difference.

This group of papers on reheat is really excellent, reflecti[ng] great credit on the authors and their respective compani[es]. The entire industry is in debt to them. To one who has bee[n] designing reheat plants off and on since 1929, these papers a[re] most illuminating in presenting so many excellent curves. The[re] would have been many more reheat plants today had the[se] papers been presented a few years ago.

I. G. McCHESNEY.[9] Mr. Parker (3) has presented an excelle[nt] review of the history of the resuperheating cycle. Discussi[on] of recent designs is of particular interest because so many opera[tors] lately have given them a great deal of careful study. Im[provement] in boiler and turbine designs are as much responsib[le] for the acceptance of the cycle as the factor of increasing co[al] costs. Often the cycle proves economical at coal costs considera[bly] below present prices.

The writer's company is installing a 50,000-kw resuperheatin[g] machine at Russell Station, having steam conditions of 145[0] psig 1000 F initial temperature, and reheat to 1000 F. Th[e] machine will operate at 1 in. Hg back pressure and have a maxi[mum] load net plant heat rate of 9622 Btu per kwhr. Studies o[f] plant economy show this machine to be 5.3 per cent better tha[n] a nonreheat machine of equal size and steam conditions.

It is interesting to note the increased interdependence of th[e] boiler and turbine design for this cycle. In some of the recen[t] designs, the stage at which steam is withdrawn for resuperheat[ing] is also used as the final extraction stage for feedwate[r] heating. Such a crossover pressure is usually too high for maxi[mum] turbine economy, and the resulting final feedwater tempera[tures] are lower than desirable for maximum over-all plant econ[omy]. A solution for this problem is the selection of a final ex[traction] stage of feedwater heating of somewhat higher pressur[e] than the stage used for resuperheating. In this case there is n[o] compromise in design, the resuperheating stage being selecte[d] for greatest turbine economy and the final extraction stage se[lected] for maximum over-all plant economy.

Fifty-six installations of various pressures totaling ove[r] 3,000,000 kw have an average maximum feedwater temperatur[e] approaching the saturation temperature within 140 F. Thi[s] approach is somewhat variable with several installations havin[g] an approach above 160 F and others near 100 F. Among thi[s] latter group are installations at Oswego, C. R. Huntley Sta[tion], Southwark, and Riverside. With small economizers usuall[y] chosen in late designs, it will be possible to obtain high econom[y] using extraction conditions for feedwater heating as shown i[n] the following table:

Throttle pressure, psig	Highest extraction pressure, psia	Turbine exhaust to reheater, psia	Final[a] feedwater temperature, deg F
1450	395	300	475
2000	825	330	520

[a] A 5-deg negative terminal difference is assumed at the outlet o[f] the final heater.

Maximum feedwater temperatures in the foregoing tabl[e] approach saturation within 118 F.

R. SHEPPARD.[10] Mr. Robertson's paper (1) is a noteworthy contribution to the published record of operating data and experi-

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[10] Assistant Division Engineer, Steam Turbine Engineering Division, General Electric Company, Schenectady, New York, N. Y. Mem. ASME.

ence on several reheat plants, and outlines the views of the author's company with regard to the controls and safety features believed necessary and desirable for steam turbines of one manufacturer. The author presents tests and performance of an early system consisting of a single intercept valve plus a steam-unloading valve. He also describes the double intercept-valve system and steam-unloading valve which is being provided on the new installations under construction. Finally, he presents for consideration a suggested scheme of two steam-unloading valves with no intercept valves at all.

Mr. Robertson states in connection with Figs. 17 and 20, that the arrangement of a single intercept valve on the downstream side of the reheater, plus a steam-unloading valve, would satisfactorily enable a reheat unit to drop maximum load without tripping the emergency governor, while the same unit with only the single intercept valve in service would trip the emergency governor. While this situation undoubtedly was present on a particular design of reheat turbine, it should not be inferred that the single intercept-valve system, if applied to other reheat-turbine arrangements, could not perform satisfactorily. All of the 28 reheat units in service listed by E. E. Parker (3) are designed with the single intercept-valve system, and satisfactory load-dump tests have been made on them.

Since a large number of reheat plants are to be built in the next few years, it is to the interest of the power industry that the turbine generators be as simple as possible to operate and low in cost to build with adequate consideration of course for overspeed safety of the turbine. The great simplicity of the (*a*) "single intercept-valve system," compared to the (*b*) "single intercept valve plus steam-unloading valve," or the (*c*) "double intercept valve plus steam-unloading valve," deserves review and examination, and might properly be presented in this discussion.

The diagram in Fig. 2 of this discussion is essentially the same as Fig. 17 in Mr. Robertson's paper (1). The intercept valve 7, and the steam-unloading valve 6 undoubtedly are necessary and essential for a turbine of the type illustrated, which has the high-pressure section and the reheat section in the same turbine casing, with high-pressure-section exhaust pressure on one side of a diaphragm, and reheat inlet pressure on the other side. This is especially true for a turbine with drum-type rotor with the reheat-diaphragm-gland diameter moderately large, and having a relatively short length of labyrinth gland with few throttlings. The entrained steam, trapped in the reheater and in the lines

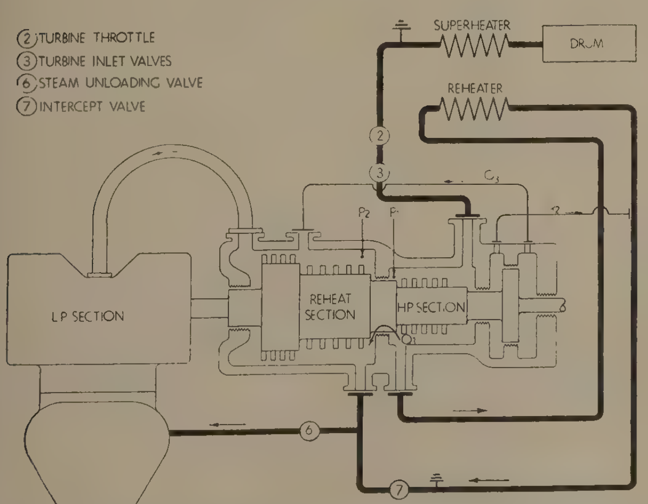

FIG. 2 REHEAT-TURBINE ARRANGEMENT WITH SINGLE INTERCEPT VALVE PLUS STEAM-UNLOADING VALVE

to the intercept valve 7, in general, contains energy of sufficient amount to produce 30 to 35 per cent overspeed. If no steam-unloading valve 6 were used, a substantial portion of this potential overspeed may be developed from backflow from the reheater to the high-pressure-section exhaust, and thence through the diaphragm gland, and then through the remaining stages.

For example, on a typical design of reheat turbine, pressure P_1 may be on the order of 450 psig. The flow Q_1 through the reheat diaphragm gland with 450 psig on one side and vacuum on the opposite side may approach or exceed the no-load flow required to drive the set. Thus upon sudden loss of maximum load, even though the main inlet valve 3, and intercept valve 7 were to close infinitely fast, this backflow may cause speed to continue to rise through the tripping range of the emergency governor. Normally, diaphragm-pressure drop (P_1-P_2) at full load may be only 50 psig or 10 to 12 per cent of the high-pressure-turbine exhaust pressure, and hence this packing leakage Q_1 normally is relatively small. However, on sudden loss of generator load (P_1-P_2) may jump instantly to 450 psig.

In addition to the foregoing leakage through the reheat diaphragm developing energy to drive the rotor after all valves are closed, the author points out that an additional source of leakage, tending to increase speed, is the high-pressure-unit balance-piston leakage Q_3. While this quantity of steam Q_3, is quite large, the energy it may develop tending to increase speed of the unit depends somewhat upon whether this leak-off steam is reintroduced into the lower stages of the turbine, or condensed in one of the feedwater heaters.

The steam-unloading valve 6 on this turbine arrangement also serves to blow down the stored steam entrapped between intercept valve 7 and the reheat-section nozzles. The large-size steam-unloading valve is a major additional capital expense. It is a valve which must be kept tight, and its stable and harmonious operation with the intercept valve requires a mechanism whose accuracy and setting adjustments must be maintained.

The single-intercept-valve scheme, Fig. 3 of this discussion, has been carefully analyzed as to its ability to drop maximum load from an 80,000-kw 3600-rpm tandem-compound unit without tripping the 110 per cent emergency governor, and a description of its features is included primarily to illustrate its simplicity and practicability when applied to the arrangement of reheat turbine being built by the manufacturer represented by the writer.

The basic essentials of the controls for this type reheat turbine, which enable it instantly to drop maximum load without the speed overshooting enough to trip the emergency governor, are as follows:

1 Rapid and early closing of the intercept valve; $^1/_4$ sec or less.

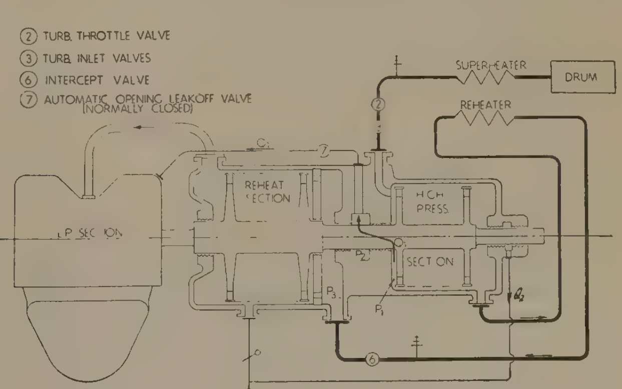

FIG. 3 REHEAT-TURBINE ARRANGEMENT WITH SINGLE INTERCEPT VALVE ONLY

2 Rapid closing of high-pressure-turbine inlet valves; 0.4 to 0.5 sec.

3 Tight-seating intercept valve.

4 Proper design of leakage gland at the reheat diaphragm and correct disposition of the bottled-up steam in the high-pressure section, so no leakages re-enter the turbine below the reheat point.

5 Location of the intercept valve on the reheat outlet just as close to the turbine as is physically possible; extraction nonreturn valves close to the turbine, to minimize entrained steam storage below the reheat points.

The impulse-type reheat-turbine arrangement (Fig. 3 herewith), with relatively small shaft-packing diameters and a large number of labyrinth throttlings at the gland of the reheat diaphragm, lends itself admirably and effectively to the simplicity of the single-intercept-valve system.

The advantages of the high-pressure inlet steam being introduced midway in the high-pressure shell on one side of the reheat diaphragm, and the reheated steam on the opposite side of this diaphragm have been discussed by E. E. Parker (3) in so far as uniformity of temperature, etc., is concerned. This design is also favorable to the speed-control problem, and can best be illustrated with a few typical pressures. Pressure P_1 at maximum load on a typical 1450-psi installation may be 1100 psig, and P_3 is 400 psig, which equals a 700-psi pressure drop across the reheat diaphragm with a subsequent large number of labyrinth throttlings required to minimize the leakage effectively.

The instant following the opening of the generator circuit breaker with subsequent closing of intercept valve 6, and control valve 3, and opening of small intermediate gland leak-off valve 7, the pressure P_1 will have dropped to 450 psig (safety-valve setting), and P_2 is less than atmospheric pressure. It is clear that with this turbine arrangement the pressure drop across the gland, originally 700 psig, has decreased instantly to approximately 450 psi with proportionally reduced leakage, which is almost totally diverted to the condenser, thus preventing the leakage flow Q_1 from producing any power.

The foregoing arrangement effectively bottles up the reheater stored steam, and totally diverts the most "critical" leakage Q_1, which could produce driving power were it allowed to expand through the reheat and low-pressure sections of the turbine. Leakage flow Q_2 from the intermediate leak-off of the exhaust of the high-pressure section may be prevented from back flowing in at the extraction stage by connecting to the heater side of the nonreturn valve. However, the connection can well be made in the conventional manner on the turbine side of the nonreturn valve, since the power, which can be generated by Q_2, even under emergency conditions of greatly enlarged labyrinth clearances, is still less than the 3600 rpm running losses of a hydrogen-cooled generator with field current off.

On several piping arrangements studied by the writer, with the intercept valve 6 placed physically as close to the turbine as possible, the stored energy in the pipes between the intercept valve and the reheat stage will produce overspeed on the order of 1 to 2 per cent. This energy is in no sense a dangerous quantity, yet it does approach the permissible limit which can be tolerated even with the most rapid types of governing mechanisms. If the turbine manufacturer is to meet the usual contract requirement that the unit must not trip the 110 per cent emergency-governor setting on loss of maximum generator load, it is essential that foundation designs permit the intercept-valve location to be given primary consideration. The importance of locating the extraction-line nonreturn valves close to the turbine has been emphasized by J. W. Barnett,[11] in connection with peak speeds of nonreheat units when dumping maximum load. In the case of reheat units, this factor is equally important.

The author states that on new reheat stations a second intercept valve 4 on the exhaust of the high-pressure section is being provided in addition to the steam-unloading valve. The purpose of this second large intercept valve is not clear. Such a valve 4 will not bottle up the steam in the reheater and prevent its flow through the lower stages in case valve 5 should stick open. Although valve 4 would prevent backflow of the bulk of the stored steam in the reheater, no benefit results from preventing this backflow through the reheat-diaphragm-gland labyrinth, because presumably it would be short-circuited to the condenser anyway through valve 8.

This second intercept valve adds major capital cost to the plant, introduces another $^3/_4$ to 1 per cent pressure drop in the reheat line, and complicates the speed-governing mechanism.

The rather novel scheme of omitting the intercept valves entirely may have merit if applied to slowly accelerating 1800-rpm units if the steam-unloading valves were made very large. Rough calculations indicate that an 80,000-kw tandem-compound 3600-rpm unit, accelerating at 12 per cent per sec, would require one steam-unloading valve at least 27 in. in diam, and a short pipe to the condenser of equal size, in order to limit successfully the peak speed to 110 per cent upon sudden loss of maximum load. Such a large 400-psi valve would have to be single-seated to insure tightness, and certainly presents some mechanical problems in order to open this valve in a small fraction of a second. The author also mentions the uncertain hazards involved in exploding the stored 1200 to 1500 cu ft of 400-psi steam into the condenser.

Two additional disadvantages of omitting the intercept valve are: (*a*) the elimination of the braking power or high rotation losses in the high-pressure element instantly after the generator load is disconnected; and (*b*) the use of a tight-seating intercept valve provides a convenient means for hydrostatic testing the reheat section of the boiler.

Such a scheme appears to be a backhanded or indirect method of shutting off the steam to the low-pressure elements, when a much smaller 18-in-diam intercept valve, placed in the reheat line at the expense of less than 1 per cent of 400-psi pressure drop, will do the same job directly.

The writer wishes to endorse heartily the author's warning that plant designers must be on the alert to see that no steam drains or by-pass lines are introduced which will put steam inadvertently into the low-pressure stages of the turbine. Relatively small quantities of steam are critical and may render the speed of the set uncontrollable.

H. A. Wagner[12] and H. E. Macomber.[13] Those interested in the design of reheat boilers, and the economics of application of reheat to utility steam power-plant cycles will welcome the detailed treatment of the subject as presented in Mr. Frisch's paper (5). His Fig. 3, showing the steam-generator surface make-up and price for both resuperheating and nonresuperheating boilers, and Fig. 5 on approximate price per kilowatt for various pressure-temperature conditions were particularly interesting. The wealth of data submitted merits more careful study than the writers were able to devote to the paper owing to the limited time before presentation. However, a further explanation of a few items would be appreciated.

The boiler designer doubtless will find Figs. 2 and 3 of paramount importance, but the user is still vitally interested in what

[11] "Speed Governing Design Consideration of Condensing Turbines," by J. W. Barnett, presented at the October, 1948, meeting of the Power Division, AIEE, Technical Paper 48-280.

[12] System Engineering, The Detroit Edison Company, Detroit, Mich. Mem. ASME.

[13] Production Department, The Detroit Edison Company. Mem. ASME.

it costs to do the job. The cost comparison in Fig. 5, it is assumed, is based on a superheater having a combination of radiant and convection surface, and a convection-type resuperheater as represented by the solid lines, since no steam-generator costs are indicated in Fig. 3 for the resuperheating cycle A_r with full convection-type surface at a gas temperature of 2000 F. For a steam generator for 100,000-kw capability, the cost per kilowatt in Fig. 3 is about the same for the 2000 psi-1050 F cycles A_n and A_r, but for the 1450 psi-1000 F cycle, the cost per kilowatt is less for the resuperheating cycle B_r than for the nonresuperheating cycle B_n. In Fig. 5, however, the reverse is true with respect to cycles B_n and B_r. Does this result from inclusion in Fig. 5 of delivery and erection or other costs not covered in Fig. 3, which reverses the trend of Fig. 3, or has it been assumed for the 1450 psi-1000 F cycle that the full-convection type of superheater was selected?

Also, if a superheater and reheater of the convection type only were considered for a furnace-exit temperature of 2000 F, it would appear from Fig. 3 that there would be considerable advantage economically to the nonresuperheating cycle B_n, as compared to cycle B_r, where the superheat control range to 0.7 full load is maintained. For the 2000-psi cycle, no comparison is possible for the full-convection superheater and reheater because the author indicates it is impractical to build a resuperheating steam generator with a full-convection superheater and reheater for a furnace-exit temperature of 2000 F. It would appear, therefore, that considerable care should be exercised in selecting the basis of comparison between boilers for reheat and nonreheat cycles.

In Fig. 3, again, it is noted that for a furnace-exit temperature of 2000 F, the amount of boiler surface B for the 1450 psi-1000 F resuperheating cycle, B_r is about 10,000 sq ft, as compared to about 23,000 for the nonresuperheating cycle B_n, or less than 50 per cent. Since the reduction in steam flow for a reheating versus a nonreheating 100-mw turbine is only of the order of 15 to 20 per cent, are these two steam generators strictly comparable?

This brings to mind the question of reliability and maintenance. Corresponding to the reduction in boiler heat-absorbing surface in the resuperheating steam generator mentioned in the foregoing paragraph, there is an addition of over 20,000 sq ft of reheater surface. Not only is this more expensive surface, but the incidence of trouble and maintenance is likely to be greater for superheat surface. In addition, higher furnace-exit-gas temperatures apparently are required in reheat-boiler design with possible increased materials troubles. Have sufficient operating data been accumulated with resuperheating steam generators of modern design and under various operating conditions to indicate that reliability and maintenance of resuperheating steam generators are on a par with the nonresuperheating boiler?

The writers again wish to express their indebtedness to Mr. Frisch for his detailed exposition on the design of reheat boilers, and the indication of the many variables involved in boiler design for resuperheating service.

Closure of Paper by C. A. Robertson (1)

Before commenting upon the several excellent papers and discussions of this Symposium, this author wishes to incorporate for the record, his opening remarks on the history of reheating.[14] These remarks were given preceding the author's main paper (1) and were requested by G. B. Warren as a suitable prelude to the papers and discussions.

Brief Review of Reheating

The thermodynamic advantages of reheating steam were recognized and applied since the early days of the steam engine. The first known patent describing a method of steam-jacketing cylinder walls to prevent condensation on inner cylinder walls was obtained by James Watt in 1769. Perhaps the first to use reheating by means of live-steam coils in the receivers of compound engines was John Bourne, who tells us he employed it in 1859.

By the turn of the century, many compound engines, principally pumping engines, were equipped with reheaters. Published tests even describe the performance of an air compressor with four steam cylinders having three stages of reheating as well as four stages of regenerative feedwater treating. It appears that reheating up to this time was done by live steam.

In 1891 Parsons devised and patented a process of interstage live-steam reheating in steam turbines by means of cored spaces in the diaphragms of a radial-flow turbine. By 1900, a 1000-kw Parsons tandem-compound turbine was tested experimentally with externally fired reheat between cylinders. A 3000-kw Parsons cross-compound geared turbine-generator unit employing both reheating and regenerative feedwater heating was built in 1916 with reheating achieved by a waste-heat boiler.[15] Occasional development and application of reheating was conducted by European engineers up to the early 1920's; however, only a small amount of performance information was published.

The year 1924 saw installation of 40,000-kw and 60,000-kw regenerative-reheating units for steam at 600 psig and 725 deg F. These turbines were, respectively, single cylinder with reheat between stages and cross-compound with reheat between cylinders. Both installations employed separate reheat boilers.

High-pressure reheat installations of the 1300-psig type were first put into service in 1925 and 1926. Exhaust steam in these installations was resuperheated in the reheater, an integral part of the high-pressure boiler, for further expansion in low-pressure units.

About 1930 both cross and tandem-compound turbines of both 600 and 1300-psig classes, were in operation or being installed with boiler reheat, steam reheat, and combinations of both. In some tandem units, the reheating was between stages of the high-pressure turbine. The first of several installations with the unit arrangement of boiler and turbine in the 1300-psig class went into operation in 1935.

The advancement of metallurgical knowledge and experience in the 1930's, permitted increases to 1300 psig without exceeding recognized limits of moisture at the exhaust. This accomplished one of the factors that led to reheating, and resulted in the installation of a large number of nonreheat plants.

Opinion was divided between the use of a simple plant without reheat, or the continuation of reheat with its higher thermal efficiency and cost. However, with the advancing cost of fuel and the advent of still higher pressures and temperatures, the reheat cycle is again gaining favor.

It is noteworthy that reheat turbines, incidentally, have been used successfully for ship-propulsion service as well as for central-station service.[16]

Closure

In the introduction to his paper (1) the author mentioned that operating experience indicates an equal flexibility in performance

[14] These remarks in elaborated form appear in *Mechanical Engineering*, vol. 71, June, 1949, pp. 504–506 and are taken from Allis-Chalmers *Electrical Review*, first quarter, 1949.

[15] "The Development of the Parsons Steam Turbine," by R. H. Parsons, Constable and Company, Ltd., London, England, 1936, pp. 19, 55, 139 and 149; also *The Engineer*, London, England, issues from January 5, 1934, to June 22, 1934.

[16] "Marine Steam Turbine Design for High Pressure High Temperature Service," by R. C. Allen, presented at a meeting of the Metropolitan Section, The Society of Naval Architects and Marine Engineers, New York, N. Y., April 28, 1944; also "Propulsion Steam Turbines for Great Lakes Vessels," by R. C. Allen, Great Lakes Section of The Society of Naval Architects and Marine Engineers, Milwaukee, Wis., May 20, 1947.

of the reheat units as compared to nonreheat turbines. He stated, "Operating records of these regenerative-reheat steam turbines have included base load and varying load operation, with frequent start-up and shutdown performance."

In reference to Dr. Christie's question as to how well reheat units will operate under frequent starts and stops and often large load swings, when modern base-load reheat units are used for varying load service, it may be of interest to discuss the experience just mentioned on reheat turbines built by the author's company. This also applies to Dr. Christie's comment relating to overheating of the exhaust at light loads.

The experience referred to in the paper has been on reheat turbines in two stations; one using separate reheat boilers to operate two turbines of 65,000 kw and 115,000 kw, respectively, installed in 1930 and 1931; the other, in which two 80,000-kw units installed in 1935 and 1943, respectively, are served by unit boilers with integral reheaters. These units are of one general type, namely, tandem-compound, with the high-pressure and reheat elements contained in the high-pressure turbine cylinder, the reheating taking place between stages in the center of this cylinder. The steam flow is unidirectional for both high-pressure and reheat elements. The exhaust from the reheat element passes to the double-flow low-pressure cylinder.

In the older station, both units were used for both base-load and varying load service, also involving long and short shutdowns and frequent start-ups. This was accomplished using separate reheat boilers, with the same degree of flexibility as for nonreheat units in the same station. Initial steam conditions, however, were 650 psig, 750 F, and the exhaust pressure was 1 in. Hg absolute. Predicted high exhaust temperatures did not occur because of the factor of relatively large heat absorption in the metal parts of the exhaust area.

In the newer station with 80,000-kw reheat units of the one-boiler-per-turbine type, with steam conditions of the order of 1250 psig, 850 F at the throttle, 850 to 900 deg F reheat, and $^1/_2$ in. exhaust pressure, the operating records show performances which include base-load operation, varying load from full load to $^1/_4$ full load, and also, periods in which frequent shutdowns and start-ups were necessary due to load conditions. This station operates in a system where, over week ends, its 80,000-kw reheat units not only must carry the base load, but also the regulation and the swings.

The subject of overheating the exhaust end of the turbine during starting, shutting down and light-load operation, based upon equilibrium exhaust temperatures, becomes more important with increases in levels of initial and reheat steam temperatures.

This subject was brought up in Mr. Parker's paper (3) and was followed by comments in the discussions by Professor Christie, Mr. Drewry, and Mr. Fiala.

Actually this subject is not so troublesome as it may appear when the one-boiler-per-turbine arrangement is used in connection with reheat cycle because of the flexibility in firing, a necessary characteristic for reheat-turbine operation. The initial and especially the reheat steam temperature can be reduced, together with the pressure, as is desired during the shutting down and starting-up periods, and also during very light-load operation, especially below first wide-open-valve load.

Operating details of this problem of controlling exhaust-steam temperatures are given by Mr. Drewry in his discussion.

During shutdowns of the 80,000-kw reheat units, operating with 850 F and 900 F full-load reheat steam temperature, the data obtained showed exhaust-steam temperatures corresponding to saturation-steam temperatures and were less than 80 deg F. This was accomplished by firing the reheat boiler to reduce the reheat steam temperature to about 700 F. This affords an easy means of meeting the desired exhaust-steam temperatures.

Emergency shutdowns, with short shutdown periods, have not been troublesome in keeping the exhaust-steam temperature within safe limits. Plants designed for high vacuum make it easier to maintain low exhaust-steam temperatures.

In connection with the arrangement of two steam-unloading valves, instead of intercepting valve or valves, for discharging steam to the condenser following sudden load trip, the author stated that the probable drop in vacuum may amount to as much as 6 or 7 in. of mercury. Professor Christie commented that this would introduce an appreciable interval to recover the full vacuum after such an occurrence.

The author agrees that ordinarily where 6 or 7 in. of vacuum

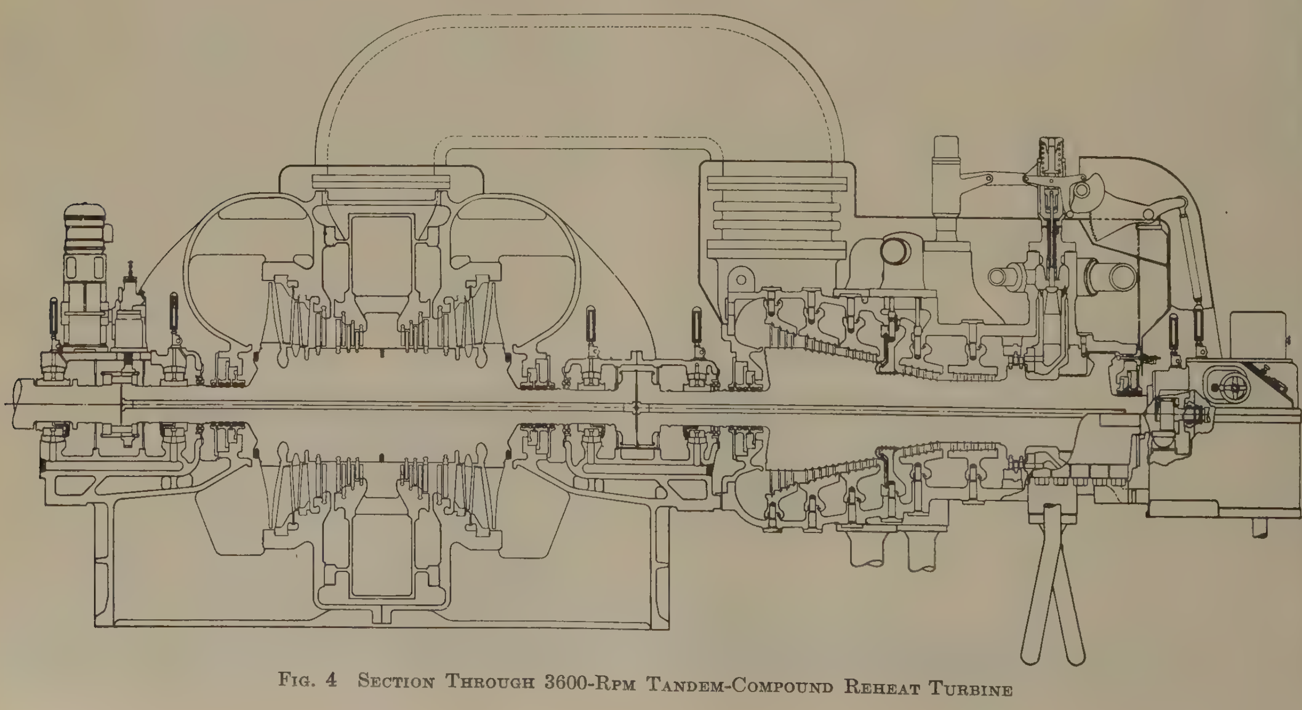

Fig. 4 Section Through 3600-Rpm Tandem-Compound Reheat Turbine

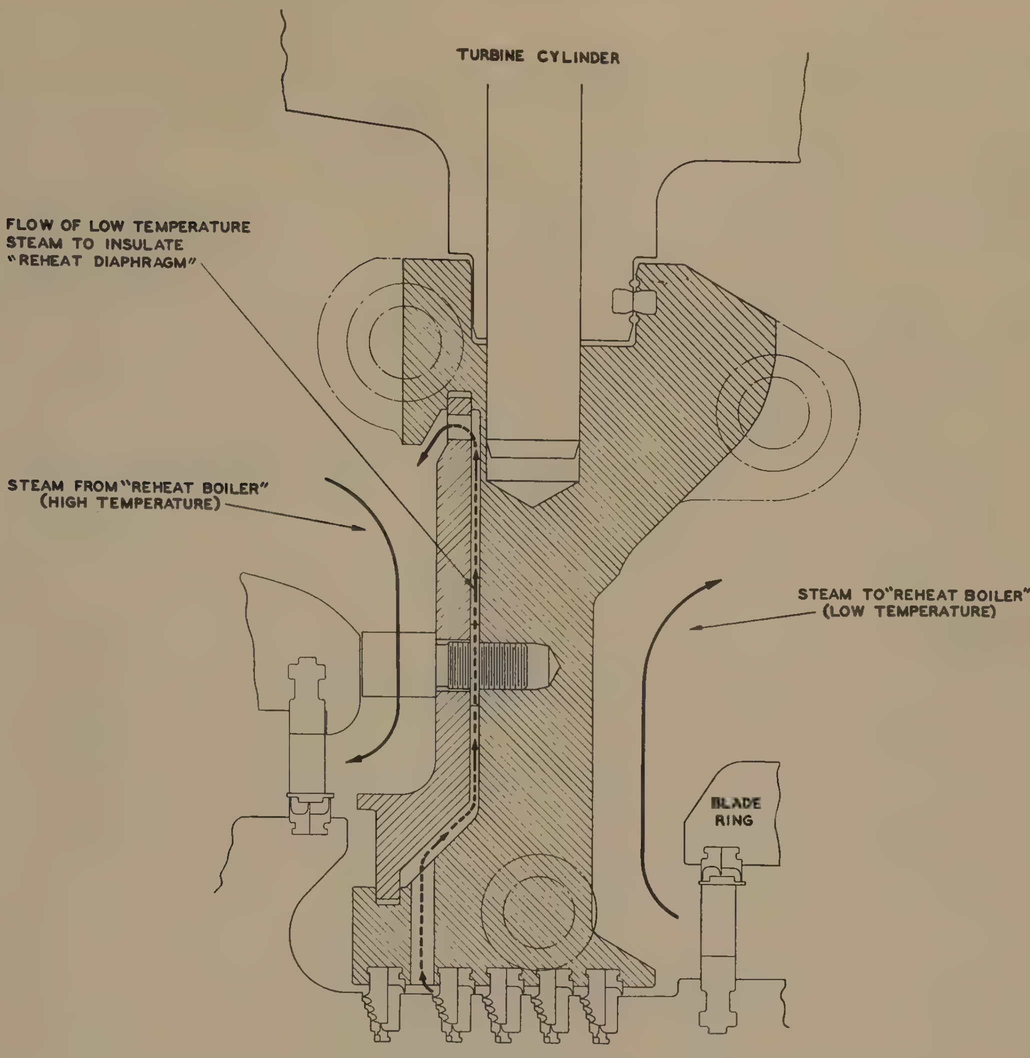

Fig. 5 Section Through Reheat Diaphragm of Turbine Shown in Fig. 4

would be lost, an appreciable time would be required to recover this loss in vacuum. The ordinary case, however, occurs because of air getting into the condenser, and reducing the rate of heat transfer, in which case considerable time may be required for the air pumps to remove this air.

In case of vacuum drop due to reheat steam-unloading valve operation, the conditions are different because the loss in vacuum is not due to air but rather due to a large terminal difference resulting from high rates of steam flow to the condenser, amounting to 200 and 300 per cent full-load steam flow. As soon as the rate falls to normal, the vacuum increases to normal without appreciable lag.

Actual unloading tests, where the instantaneous rate of steam flow to the condenser through the steam-unloading valve approached 75 per cent of full-load steam flow, the vacuum gages showed no appreciable change. The time interval or duration was less than $^1/_2$ sec, and the actual quantity of steam was not more than 200 lb.

In the case of the 80,000-kw reheat-turbine installation, the actual quantity of stored steam in the reheater and connected piping is about 750 lb of superheated steam.

The author appreciates particularily Mr. Sheppard's discussion of the merits of single-valve reheat protection, as compared to the two-valve protection provided by the author's company.

In other words, whether one, or two lines of defense are needed against overspeeding from reheat steam is basically the cause of the conflict in thinking.

Mr. Sheppard advocates a single intercepting valve, yet near the end of his discussion he admits that intercepting valves can stick open. He apparently fails to understand that some of the plans described in the author's paper provide two lines of defense.

This subject of one versus two lines of overspeed defense is covered in Mr. Drewry's discussion.

Putting balance-piston or packing leakage into an extraction heater past the check valve, as recommended by Mr. Sheppard, apparently has poor reliability and economy aspects.

On the subject of substitution of steam-unloading valves for intercepting valves, Mr. Sheppard notes two disadvantages of omitting the intercepting valve. The latter disadvantage states, "the use of a tight-seating intercept valve provides a convenient means for hydrostatic testing of the reheat section of the boiler." For an intercepting valve to be used for hydrostatic testing of the reheater, requires another valve in the reheater inlet line, for which credit could have been given in discussing the two intercepting-valve plan. Gate valves are but a minor fraction of the cost of intercepting valves.

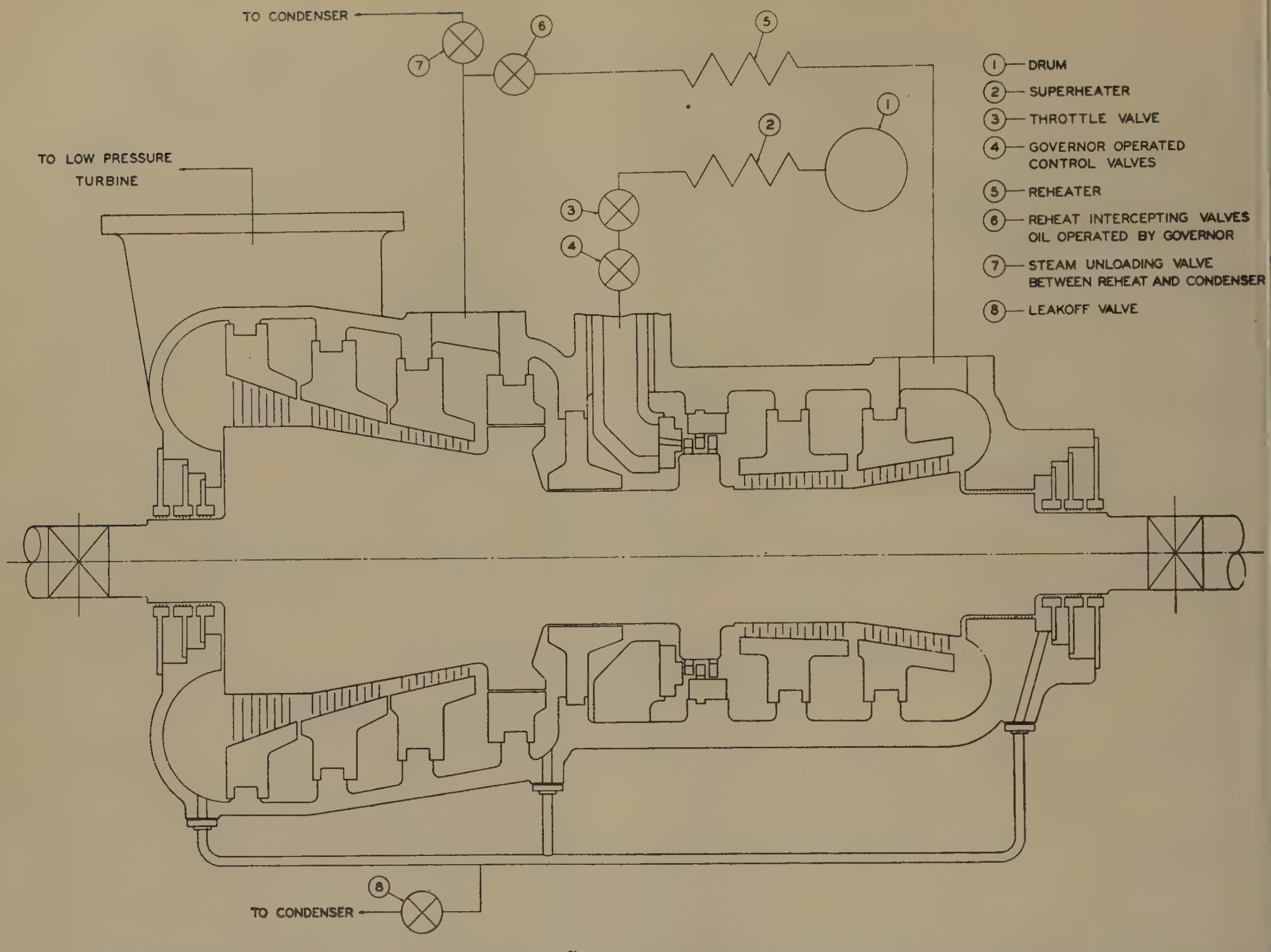

Fig. 6 Impulse-Reaction Turbine to Provide Admission Steam and Reheated Steam at Adjacent Points in Center of Cylinder

The author agrees with Mr. Sheppard's statement as to the advisability of placing the intercepting valve as physically close to the turbine as possible, although this may require expensive changes in turbine foundation. In this connection, minimizing volume of the reheat system could well have been urged, thus reducing the quantity of steam in the reheat lines and boiler.

The advantages of using a steam-unloading valve, by-passing steam to the condenser, should give an answer to the foregoing problem.

Using Mr. Sheppard's value of $^3/_4$ to 1 per cent pressure drop for one intercepting valve, and calculating its economy aspect for an 80,000-kw unit, in connection with the suggested scheme of two steam-unloading valves with no intercepting valves at all, the savings by the latter scheme would be approximately $2500 per year, based upon 75 per cent load factor and $0.33 per million Btu of steam supplied. This amount capitalized would represent an appreciable saving.

In connection with the two parallel steam-unloading valves suggested in the author's paper (1), and using certain values given by Mr. Sheppard, together with calculations by the author, the following answers can be determined:

Following load trip-out, the steam in the reheat boiler would divide in two ways, namely, about 80 per cent would by-pass the reheat section of the turbine directly to the condenser through two parallel steam-unloading valves, and the other 20 per cent would flow through the reheat section of the steam turbine to the crossover. Two valves are used instead of one so as to provide two lines of defense.

Using the values given by Mr. Sheppard in his discussion, namely, an 80,000-kw, 3600-rpm reheat unit with low mechanical inertia and an accelerating rate of 12 per cent speed rise per sec with full-load torque, the total rise in overspeed would be approximately 10.8 per cent with the two steam-unloading-valve arrangement.

Using a simple method of analysis, let us assume that $^1/_2$ sec elapses before steam-unloading valves become effective in by-passing steam to the condenser. Therefore the speed rise will have reached about 6 per cent overspeed. Assume that the unit will reach 30 per cent overspeed if unprotected by reheat-unloading valve or intercepting valves. On the basis that 80 per cent of the steam is by-passed, the following calculation indicates the speed rise to be expected after the unloading valves have functioned

$$(30 - 6) \times (100 - 80) = 4.8 \text{ per cent}$$

Total rise would be 6 + 4.8 = 10.8 per cent overspeed rise. This is somewhat in agreement with a more detailed analysis.

The author is including several figures to complete this closure.

Fig. 4 shows a section through a 3600-rpm tandem-compound reheat turbine designed along the conventional lines, being used by the author's company. This turbine has the reheat diaphragm located in the center of the high-pressure cylinder in which the

steam flow is unidirectional, and the reheating takes place between stages in the center of the cylinder.

The temperature difference at the reheat diaphragm and adjacent cylinder structure, during the starting cycle, is less in this unidirectional-flow design than in the opposed-flow design of the turbine described by Mr. Sheppard. This is due to the inherent characteristics of reheat boilers. As the reheat boiler is being brought up, the temperature rise through the reheating element is low.

During normal operation, the difference in steam temperature between the two sides of the reheat diaphragm is the highest and ranges between 250 and 300 deg F. The temperature gradient, however, is broken up between two gas films adjacent to the metal, and the metal of the diaphragm itself. The temperature gradient through the films is relatively high leaving a considerably smaller gradient through the metal. Units that are in operation at the 850 deg F level operate with a temperature difference at the reheat up to 260 deg F, as compared to 320 deg F for a 2000-psig, 1050 or 1000 deg F reheat installation.

Another way to reason that this temperature gradient is not of a serious nature is to compare it with the temperature gradients experienced in existing nonreheat units operating throughout the country.

Consider the temperature gradient of a nonreheat turbine rotor at the high-pressure end where a water-seal gland runner surrounds the shaft, and where the oil-cooled No. 1 bearing and journal are located. In a relatively short distance there is a very large temperature gradient from 900 to 1000 deg F to around 200 deg F, yet not very much concern has been made of this fact.

In the high-pressure end of a standard steam turbine there exists usually a large temperature gradient between initial steam temperature and a relatively cool temperature of the metal at the extreme end of the turbine cylinder.

Fig. 5 shows a sectional view of the reheat diaphragm for the turbine shown in Fig. 4. It will be noted that the diaphragm is of ring construction, bolted together at the horizontal parting joint. This type of construction provides freedom for radial expansion within the turbine cylinder. This freedom for radial expansion is similar to that provided in the 80,000-kw reheat units referred to in the author's paper (1). These diaphragms have a small radial depth.

The particular diaphragm shown in Fig. 5 was designed for high-temperature service in that a special shield is provided on the low-pressure side. The leakage steam from the cooler and higher pressure side of the diaphragm, after passing a series of labyrinth seals, enters the space between the shield and diaphragm and flows radially to the circumference where it mixes with the reheated steam. By this means the two sides of the diaphragm are kept at substantially the same temperature which minimizes any tendency to warp. The larger temperature differences occur over the relatively thin shield which has provision for radial expansion.

This view in Fig. 5 also shows the large number of throttlings which are provided for in what is termed a relatively short diaphragm by Mr. Sheppard, and which was also a characteristic of the diaphragms for the 80,000-kw reheat turbines mentioned in the author's paper (1).

Fig. 6 is introduced to show how an impulse-reaction turbine would have to be built in order to have the admission steam and reheated steam at adjacent points in the center of the cylinder.

In order to take care of all operating conditions, such as, starting, normal-load operation, and the condition following unloading where the intercepting valve is closed, our study indicates that, in order to balance the steam thrust axially, a separate balance piston is required for each element.

This type of turbine also requires much more elaborate shaft-seal packing at the high-pressure end where the shaft passes through the cylinder and at the reheat diaphragm.

There is a rather large temperature difference at the center of the turbine during the starting period where low reheat temperature exists. This design does nothing toward simplicity, neither in design nor for operation, and the resulting differential expansions, both radial and axial under all operating conditions, are not less than that for the unidirectional-flow design used by the author's company.

Closure of Paper by E. E. Harris and A. O. White (2)

Calculations have been made to check Mr. Caldwell's statements, and we are unable to determine that it will be possible to realize "60 per cent of the increased economy attainable with conventional resuperheating."

When light-load operation is mentioned along with the heating of the exhaust, as in Mr. A. C. Christie's discussion, reference is made to about 5 per cent of the turbine rating. Certainly, 5 per cent of rating is not usually considered as ordinary operating conditions for most installations.

Mr. Sheppard pointed out a method incorporating use of a single intercept valve, not mentioned by Mr. Christie. This method has been used by the authors' company since the first two Philo units were installed, as mentioned by Mr. Sporn.

In a future paper, we hope to present data on gains due to reheat at various loads from which economic studies may be made at lower loads.

Referring to Mr. M. K. Drewry's comments, the question of exhaust temperatures at light loads needs no further explanation. Tests run in conjunction with the American Gas and Electric Service Corporation, as mentioned by Mr. S. N. Fiala on the 2300-lb reheat unit at Twin Branch, showed no evidence of overheating. However, calculations on 3600-rpm units with the higher tip speeds indicate that cooling will be required on these units. Mr. Drewry is correct in stating that the exhaust can get too high in temperature even on nonreheat units at very light loads. The exhaust temperature on starting and for light-load operation may be controlled readily by means of water sprays automatically controlled. His comments on possible troubles with desuperheating are interesting.

Mr. J. N. Ewart in his discussion appears to be fully acquainted with all the problems associated with reheat for modern steam stations and feels that the reheat station has come into its own for the present and for some time to come.

Mr. S. N. Fiala has been interested in the design, construction, and operation of reheat stations for many years. His frank opinions should be reassuring to those who have more recently taken on reheat installations.

In reply to Mr. R. W. Hartwell, it was pointed out in the paper (2) that the gain due to reheat was predicated upon the assumption that the nonreheat unit would have a kilowatt exhaust loss of 4.5 per cent and, by using the same last-stage annulus area with reheat, the reduced condenser flow would give a reduction in exhaust loss. Usually the smaller-rated units are not so crowded in the exhaust end, consequently do not benefit as much from reduction in exhaust loss for the reheat unit over the nonreheat unit.

The efficiency of a unit depends considerably upon the high-pressure end volume flow. The smaller-rated capacity units have a lower efficiency owing to lesser volume flow, so that any reduction in volume flow in the order of 16 per cent for the reheat units will cause a further reduction in high-pressure section efficiency. Hence, it becomes more difficult to maintain similarity of design. We feel that the paper (2) as presented will cover units rated 50,000 kw and larger, and, for units smaller than the 50,000-kw

rating, the gain obtained will be approximately 1 per cent less than the paper describes.

We concur with Mr. Hartwell's other remarks and feel particularly favorable toward more economic studies being made to obtain a better idea as to the relation of the proper reheat pressure. In order to be of assistance in such a study, we are continuing our calculations to cover the lower-load operation.

The gain due to reheating Figs. 9, 10, 11, and 12, in the paper (2), is the gain at maximum capacity plotted against reheat pressure for which the unit is designed at maximum capacity. After the reheat pressure has been selected for the maximum capacity, the reheater pressure will vary almost directly with the throttle flow. However, the gain at any reheater pressure for partial loads is different than that shown on the curves in the paper (2), Figs. 9, 10, 11, and 12. At partial loads a curve of gain due to reheat for a unit designed for a given reheat pressure will intersect a gain curve at the reheat pressure for maximum capacity and will be similar in shape, without discontinuity, at the lower reheat pressures or partial flows, and will be lower than the locus curve of the reheat pressure at maximum load.

Mr. E. H. Krieg's remarks on costs and economics of reheat versus nonreheat plants are interesting and timely. We sincerely hope that more economic studies will be made available as time goes on to help clarify the situation.

The gains due to reheating as presented in the authors' paper (2) represent the gain in the turbine room, thus including changes in turbine performance, heater performance, reduced flow to the condenser, along with change in boiler-feed-pump power. No account was taken for the reduction in circulating-water-pump power or other station auxiliaries or boiler efficiency. If the other station auxiliaries and the boiler efficiency remain the same, then the gains may be interpreted as per cent gains on the whole station heat rate.

Mr. I. G. McChesney's suggestion that the top extraction point and the reheat point be separate and each at its best pressure level has been followed in several installations now in progress. Here again some economic studies would be valuable, and his table of suggested pressures is very interesting. It is the authors' intention to continue their studies in an endeavor to be of assistance in deriving such economic data.

Closure of Paper by E. E. Parker (3)

The material presented by the various discussers constitutes a valuable addition to this symposium on the reheat cycle. This is particularly true as the discussions in large part are based on over-all station operating experience with existing reheat installations and the experience to date on the design of new reheat installations. It is reassuring to note throughout the discussions that this experience has been favorable and no serious drawbacks encountered.

A few points raised in the discussions deserve brief comments. The proposal of Mr. Caldwell for resuperheating the steam within the turbine by extraction of steam from a higher stage has been studied and the result of this study presented in a discussion at the Annual Meeting in 1947. Briefly, we are unable to agree with the gain in efficiency claimed by Mr. Caldwell even assuming adequate heat-transfer area within the turbine to allow the desired amount of resuperheating. According to our study, the gain is insufficient to justify the consideration of this scheme. Actually when the physical conditions necessary to provide adequate heat-transfer area are taken into consideration, the gains become even less.

Considerable divergence of opinion exists as to the proper way of protecting the turbine against overspeed due to the steam stored in the reheater. This is an important problem and deserves full consideration. As pointed out in the author's paper and also by Mr. Sheppard in his discussion, this has been successfully met on the large number of reheat turbines supplied by the author's company through the use of a single intercept valve. Such an arrangement gives adequate protection consistent with that provided on a nonreheat installation and provides the greatest simplicity of control. Simplicity of control is important in reducing the operational objections to the reheat cycle. It is perhaps not fully appreciated that most nonreheat installations include possibilities for re-entry of sufficient steam into the turbine from the extraction system to cause overspeed of the same magnitude as the steam from the reheater. Nonreheat installations are carefully reviewed to provide adequate protection against the re-entry of steam in the form of free-swinging or positive-closed check valves. The single intercept valve meets the same standard of protection as used in the design of such check valves. The use of the single intercept valve has none of the objections raised by Professor Christie against the methods which he describes to accomplish this function.

Professor Christie points out the operation of a nonreheat installation is simpler and for this reason may be chosen in preference to a reheat installation where loads are variable. For conditions involving frequent starting and stopping this is undoubtedly true, but it should not be overlooked that for the variations in loads usually considered on central-station systems reheat installations have proved satisfactory. For information on the satisfactory performance of a reheat installation on very rapid variations in loading, attention is called to a paper by V. F. Estcourt entitled "Design and Operating Problems With Gas and Oil-Fired Boilers for Stand-By Steam Electric Stations," ASME Transactions 1937, vol. 59, paper No. FSP-59-1.

Mr. Krieg has compared Preferred Standard turbine generator units with nonpreferred units on the basis of nameplate rating rather than on the basis of capability. The author dose not feel that this represents a true comparison.

Preferred Standard turbines have been purposely designed with capability of 10 per cent greater than the nameplate rating whereas the nonpreferred standard turbines have 25 per cent. When both the turbine capability and generator capability are taken into consideration the cost differentials between preferred standard and nonpreferred standard units would be reduced to approximately one fourth the values stated by Mr. Krieg. When compared on this basis many purchasers have found it economical to select reheating for units within the Preferred Standard range of sizes.

Closure of Paper by Martin Frisch (5)

In closing, the author of paper (5) wishes to express his appreciation and thanks to those who have taken the great trouble to prepare written discussions and to those others who have raised questions from the floor about his paper. These discussions and questions are, perhaps, more interesting and useful than the paper itself because they reveal the character of the reflective thinking applied to the estimation of the debits as well as credits of resuperheating.

It has probably escaped few who have listened to these papers on resuperheating that the difficulties and debits, real as they are, do not seem as serious to those who have installed and operated resuperheating units as to those who have just been thinking about them.

The problems of quick starting and emergency-load loss are not belittled in any of the papers or discussions. But, as brought out in discussions by Mr. Sporn, Mr. Ewart, Mr. Fiala, and Dr. Christie, the safety of the turbine seems to be of no less concern than the safety of the steam-generating unit in emergencies. In this author's opinion a resuperheating steam generator,

whether employing convection superheaters and resuperheaters or radiant superheaters and resuperheaters, can be made as safe during emergencies as units of the nonresuperheating types with little if any added complication.

Mr. Drewry's data summarizing experience with radiant superheaters and resuperheaters totaling over "500 superheater years" with negligible difficulties should give heart to those who will wish to avail themselves of the advantages of resuperheating with slagging fuels without accepting the higher furnace-exit temperatures necessary for any considerable control range with exclusively convection arrangements. The ability to handle all locally available low-cost fuels without the need to be selective is important if the economies of resuperheating installations are to be realized over their entire life, even when base-loading them ceases to be desirable. Reasonably low-priced units, employing radiant superheating surfaces, can be designed for slag-free operation with the poorest fuels.

Messrs. Wagner and Macomber in their discussion raise questions about a seeming inconsistency between prices as read from Figs. 5 and 3. The prices in Fig. 5 and Fig. 3, except for the small effects resulting from drawing smooth curves through the points, are consistent, as may be seen from Table A of this closure.

TABLE A STEAM-GENERATOR PRICE

(Dollars per kilowatt for 100,000-kw capability unit, 2000 F, gas to superheater 70 per cent control range)

Cycle	A_r	A_n	B_r	B_n
From Fig. 5	19.50	19.40	17.90	17.20
From Fig. 3:				
Combination radiant and convection superheater	19.50[a]	19.40	18.20	18.70
Convection superheater, standard firing	Impossible	19.40[a]	18.00[a]	17.25[a]
Differential firing	Impossible	18.90	17.00	17.05

[a] Fig. 5 based on figures marked with [a].

In Fig. 5, cycle A_r, prices are based on units including radiant superheaters since the required primary steam temperature of 1050 F could be maintained down to 70 per cent of full load, with no other arrangement with a gas temperature of 2000 F to the superheater. Prices for other cycles in Fig. 5, are for units having exclusively convection superheaters and resuperheaters which, with the gas temperature of 2000 F at the superheater and with standard firing, can maintain the required steam temperature down to 70 per cent of full load. For these cycles, somewhat lower prices would be possible with differential firing as may be seen from Table A. Perhaps, it might have been better if the prices plotted in Fig. 5, had been the lowest from Fig. 3, for each of the cycles. If this had been done, prices of resuperheating units for cycle B_r would have been just a shade lower than for the corresponding units for the nonresuperheating cycle B_n.

Fig. 3 shows that, for the same operating conditions and fuel specifications, it is not advisable, as suggested by Messrs. Wagner and Macomber, to compare prices only of units with exclusively convection superheaters and resuperheaters or, alternatively, of units including radiant surfaces. What is wanted is the lowest-priced unit, irrespective of the arrangement, which will meet the conditions and specifications.

The difference in the boiler surface B from Fig. 3, with 2000 F gas temperature for cycles B_r (10,000 sq ft), and B_n (23,000 sq ft) comes about from the fact that both units with radiant superheating were designed for the same unit exit-gas temperature. Therefore, 13,000 sq ft additional boiler surface had to be used in the unit for cycle B_n to compensate for the effect of the resuperheater and the smaller gas quantity on exit-gas temperature of the B_r-cycle unit. For the same reason, in the corresponding all-convection superheater and reheater units with 2000 F gas temperature, no boiler surface at all was necessary for either the B_n or B_r cycle. But it was necessary to use much more economizer surface with cycle B_n than with cycle B_r. In all cases the surfaces are based on actual steam flows required for the 100,000-kw turbine capability as shown in Fig. 1.

It is true, as Messrs. Wagner and Macomber point out, that, in resuperheating units, more metal works at high average metal temperatures than in nonresuperheating units. This, naturally, has an effect on life, reliability, and availability, but, with proper design, resuperheaters can be designed to equal superheaters in these requirements.

It was gratifying that Mr. Krieg and Mr. Rowand were both able to confirm the conclusion that resuperheating steam generators cost little if any more per kilowatt than nonresuperheating units for the same primary-steam and fuel specifications.

Mr. Gaffert's question as to whether resuperheating units could be considered economical for conditions existing in the Middle West, where many units would necessarily have to be smaller than 50,000 kw, cannot be answered from the studies on which this paper is based. This paper shows the effect of size on steam-generator price only down to 50,000 kw, and complete plant studies for the actual conditions would have to be made to determine whether resuperheating would pay.

Mr. Donworth's question as to whether the gains in efficiency from resuperheating could not be more easily matched by more regenerative feed-heating cannot be answered by this study which considered only the effects of design conditions on steam-generator prices. Mr. Donworth also questions the propriety of drawing smooth price curves for each of the cycles from capabilities of 50,000 to 150,000. He is, of course, correct in raising this question, and the curves should have steps or discontinuities. It was found that these could be smoothed out without affecting the conclusions, and this was done for convenience.

Closure of Paper by W. H. Rowand, A. E. Raynor, and F. X. Gilg (6)

The authors of this paper (6) are glad to have had an opportunity of contributing information from their reheat-boiler experience to this symposium on the reheat cycle.

Regarding the possibility of using radiant superheaters and reheaters, we would like to point out that radiant superheaters and reheaters have been built and are operating successfully. However, our observation has been that to avoid the additional operating considerations necessary to provide high operating reliability with radiant superheaters, many users prefer the more flexible convection superheaters and reheaters for the present level of temperatures. However, when still higher steam-temperature levels become practical, radiant superheaters may be used more generally.

Disregarding any theoretical considerations of the temperature of gases entering the convection section required to prevent slagging difficulties, there are dozens of boilers operating successfully without slagging trouble, using the worst midwestern coals, with actual gas temperatures entering the convection section of 2100 F to 2300 F. Of course, these jobs have widely spaced tubes, shallow tube banks, and modern mass-action soot blowers. The reheat units described by Mr. Gilg (6) are designed to the same furnace factors successfully proved in operation by these many installations.